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PASSENGER CAR SPARK IGNITION DATA BASE
Volume II: Discussion and Results

By

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16. Abstract

Test data was obtained from spark ignition production and preproduction engines at the engine and vehicle level. The engines were applicable for vehicles 2000 to 3000 pounds in weight. The data obtained provided trade-offs between fuel economy, power, emission constraints and emission control technology, and other relevent parameters.

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PREFACE

In support of the U.S. Department of Transportation, National Highway Traffic Safety Administration, Office of Research and Development, the U.S. Department of Transportation, Research and Special Programs Administration, Transportation Systems Center contracted with Volkswagenwerk AG, Federal Republic of Germany to develop a data base on passenger car spark ignition engines. Volkswagen production, pre-production and research spark ignition engine systems were used for the test portion of the work. Published and unpublished literature was used for the theoretical studies.

The report consists of three volumes. Volume I, the Executive Summary, presents a summary of the data obtained and a review of the important conclusions. Volume II, the main body of the report, provides a discussion of the fuel economy and emissions obtained, a description of the engine/vehicle systems tested and the results of factory driveability tests, Volume III, the appendixes, presents miscellaneous data used during the program.

The author wishes to acknowledge the guidance and assistance provided by Mr. H. H. Gould and Dr. R. R. John of the Department of Transportation - Transportation Systems Center, and Dr. K. H. Digges of the Department of Transportation - National Highway Traffic Safety Administration, Office of Passenger Vehicle Research.

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Misses E. Logsdon, M. Sowka, H. Welke
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They were supported by Dr. K. H. Lies and P. Seifert and their groups together with members of the staff of the VW Engine Testing Department, the VW Programmed Control System Department, and the VW Measuring Department.

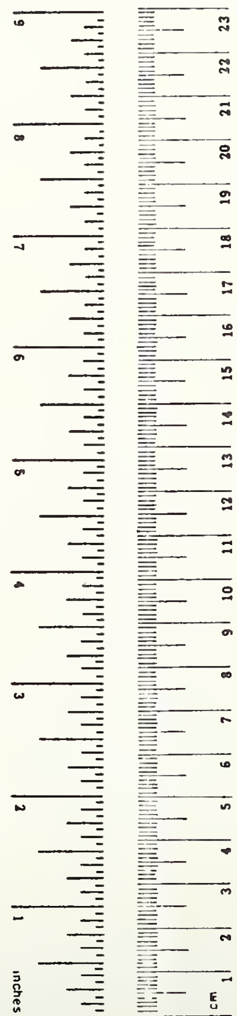
For translation work we acknowledge the assistance provided by Mr. Wilfried Becker, Germersheim, Federal Republic of Germany.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

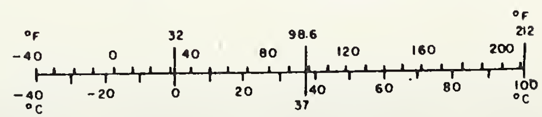
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10 286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1. SUMMARY

1.1 INTRODUCTION

1.1.1 Objectives

The objective of this effort was to obtain an engine level and vehicle level data base on the performance of a highly fuel efficient spark ignition (Otto Cycle) engine family now in pre-production or production status for passenger automobiles. The data base was for an engine family that is applicable to vehicles in the 2000 to 3000 pound inertia weight classes, which includes peak horsepower to vehicle inertia weight ratios of about 0.025 horsepower/pound. The data objective was to provide trade-offs of maximum fuel economy vs. emissions as a function of weight, power to weight ratio, engine displacement, and other relevant parameters and constraints defined herein.

1.1.2 Items of Work

The specific items of work required in this effort were to develop a program plan, a data base framework, a description of the engines formerly used, and an engine/vehicle test plan, as well as to perform all necessary engine/vehicle tests and document test data.

Program Plan

The program plan defined:

(a) Schedule and methods for obtaining experimental data and available data.

(b) Methodology to be employed in analyzing and presenting required results and trade-offs.

(c) Plan of expenditures by work items and by month.

(d) Major milestones.

Framework

The data base framework included the important design parameters and variables and showed the relationship between these parameters and how they affect fuel economy of the engine family.

The framework included, but was not limited to, the following parameters:

(a) Maximum fuel economy (EPA urban and EPA highway cycles).

(b) Emission control levels, including

1. 1.5/15/3.1 gms/mi HC/CO/NOx respectively
2. 0.9/9.0/2.0
3. 0.41/3.4/1.0
4. 0.41/3.4/0.4
5. Uncontrolled

as a function of the corresponding emission control technologies, including oxydation and reduction catalysts, proportional exhaust gas recirculation (EGR), fuel metering configuration and spark control. Feed gas composition was also considered a variable.

(c) Engine peak horsepower to inertia weight ratio down to approximately 0.02 HP/Lb.

(d) Engine CID to peak horsepower ratio.

(e) Drivetrain type and parameters.

(f) Noise and unregulated emissions (e.g., sulphates).

(g) Reliability, maintainability, durability and service life of engine, engine sstem and related vehicle systems.

(h) Acceleration performance and gradeability.

(i) Engine system weight and volume (including required accessories/auxiliaries).

(j) Startability.

(k) Driveability under varying environmental conditions encountered in the fifty states and Puerto Rico.

(l) Energy available for heating and cooling.

(m) Vibration.

(n) Fuel requirements.

(o) Requirements for non-standard materials.

(p) Electromagnetic interference.

(q) Engine system cost and other impacts.

Description of Engine Family

The description of the engine family described the engines and engine/vehicle systems, including operating characteristics, which were employed to generate the data base and trade-offs. The documentation is in sufficient detail to support the data developed under the framework, and the resulting trade-offs. System changes/recalibrations and controls shall be described.

Engine/Vehicle Test Plan

The test plan for tests of the engine family augmented data already available to meet program objectives for DOT approval. Data already available was listed. Modifications to the engine required to meet the various constraints with maximum fuel economy were specified.

Engine/Vehicle Tests

VW performed at their facility, all necessary tests at the engine and vehicle levels that would show the relationships of the parameters given in the Framework, and the maximum fuel economy for given constraints for the engine family. Engine maps and EPA urban and highway test data are provided for relevant configurations.

1.1.3 Reports

Monthly Progess Reports and this Final Report were submitted.

Monthly Progress Reports

Monthly progress reports covered activities of the preceding month. The progess reports included the essential data experimentally or otherwise obtained and reduced, and the highlights of the work accomplished.

Final Report

This final report includes all data relevant to the engine family under study; the significant impacts, projections of fuel, economy, durability, and all other items specified in the Framework, relationships between parameters, and significant tradeoffs.

1.2 SCOPE OF WORK

The scope of this 16-months study (October 1976 - January 1978) involved a number of related tasks which may be briefly described as follows.

Task I - Consisted of selecting an engine/vehicle system, in the 2000 to 3000 lbs inertia weight range with peak-horsepower-to-inertia-weight ratios between 0.02 and 0.03 in which all fuel economy figures are comparable regardless of the emission standard of the engines.

TASK II - Consisted of selecting engine systems able to meet the emission control levels:

Uncontrolled

1.5/15/3.1 gpm HC/CO/NOx respectively

0.9/9.0/2.0

and the engineering goals of the emission control levels

0.41/3.4/1.0 gpm HC/CO/NOx respectively

0.41/3.4/0.4

Task III - Consisted of defining the engineering goals of the emission control levels 0.41/3.4/1.0 and 0.41/3.4/0.4 without sufficient experience of durability and field tests involving suitable engine concepts.

Task IC - Consisted of developing emission control systems for the engineering goals of the emission control levels 0.41/3.4/1.0 and 0.41/3.4/0.4 with optimum fuel economy in each case of engine/vehicle system.

Task V - Consisted of startability and driveability testing under US-Federal conditions with the optimized engine/vehicle systems as mentioned above and deciding if they are acceptable or not.

Task VI - Consisted of collecting the main engine map data of the optimized engines which passed startability and driveability tests.

TASK VII - Consisted of compiling the passing ability and gradeability of the optimized and acceptable engine/vehicle systems using their engine map, transmission and vehicle data.

Task VIII - Consisted of evaluating the noise figures and the hydrogen cyanide and sulfate emissions of the engine/vehicle systems considered.

Task IX - Consisted of evaluating the cost figures of the engines including their different control systems.

Task X - Consisted of a fuel economy impact investigation and evaluation of the most important vehicle, transmission and engine data.

1.3 MAJOR CONCLUSIONS

The data of vehicles and engines which were evaluated under this contract are listed in Tables 1.3.1 through 1.3.3 summarizing the major test results.

1.3.1 Fuel Economy

Fuel economy is affected mainly by inertia weight, air drag, drivetrain, engine displacement, peak horsepower and emission standard.

Reducing engine displacement by 20 % within acceptable consumer attributes improves fuel economy by 5%.

Reducing peak horsepower by 15% within acceptable performance, improves fuel economy by 5%.

The composite fuel economy of vehicles in the 2250-3000 lbs inertia weight range equipped with 4 and 5-cyl. naturally aspirated spark ignition engines varies by 78% from 18 to 32 mpg.

Within the range of 4-cylinder engines alone a scatter bandwidth of 35% can be found.

4-cylinder 1.6 liter engines consume between 0 to 12% more fuel than comparable 1.3 liter 4-cylinder engines (same inertia weight and emission level).

4-cylinder 1.6 liter engines producing 85 hp use 1% more fuel than comparable 4-cylinder 1.6 liter engines producing 67 hp (same inertia weight and emission level).

Emission Level

Changing the emission level from uncontrolled to 1.5/15/3.1 and 0.9/9/2.0 respectively (gpm HC/CO/NOx) causes fuel economy to drop by between o and 12%.

Disregarding the change from 4-cyl. to 5-cyl. engine the introduction of the fuel injection (K-Jetronic) with closed loop necessary to attend the 81 Federal Standard leads to an increase in fuel economy of 4 to 5%.

This tendency is illustrated in Figure 1.3.1 by the example of the 1.6-liter Rabbit engine and the inertia-weight class of 2,250 lbs. The first three bars under each Modification Code show the declining emissions, while the last bar represents the combined fuel economy. A slight drop in fuel economy is already evident at 1976 conditions as opposed to uncontrolled conditions. The 1981 Federal Standards lead back to the uncontrolled level at substantially higher initial costs. However, the fuel economy again decreases at a NOx level of 0.4 gpm.

In other words, a comparison with present production vehicles indicates that the fleet fuel economy may be expected to improve by 4% or 5% if a closed-loop system is introduced and a stoichiometric air/fuel ratio can be maintained in the field. The improvement would be lost immediately, however, as soon as the NOx emission standard is lowered to 0.4 gpm.

Vehicle Weight

The mass of a vehicle is composed of its inertia weight and its payload. A high payload means high absolute fuel consumption. Nevertheless, for reasons of fuel economy it would be desirable to have higher payloads. The higher the payload, the lower the percentage of fuel used for moving the inertia weight of the vehicle, or, in other words, fuel economy is much improved if six persons travel not in six passenger cars but in one.

The only factor which manufacturers of automobiles can influence is the inertia weight of a vehicle. From the aspect which is of interest here, Fig. 1.3.2 shows the link between inertia weight and fuel economy. A closer look shows that the relationship is not linear, because the influence of a constant absolute variation of the inertia weight necessarily loses significance as the size of the inertia weight itself grows.

Based on our studies, we are in a position to state that increasing the interia weight from 2,000 to 3,000 lbs will result in a fuel economy loss of between 8 and 11%, provided that all other parameters and especially engine size and type are kept constant. In

the field, fuel economy losses resulting from inertia weight which increases from 2,000 and 3,000 lbs are closer to 20 rather than to 10%, because field tests involve bigger engines as well.

Performance

Figure 1.3.3 shows fuel economy versus horsepower-to-inertia-weight ratio. The vehicles under investigation were not equipped with special emission control systems. All figures are combined results.

It is discernible that in the range from 0.02 to 0.04, which is of special importance under the terms of this contract, the horsepower-to-inertia-weight ratio without specifying weight has no influence at all, as practically any fuel economy can be allocated to each of the figures given. Of those factors which do have influence, inertia weight is foremost among them. We can see that any change in inertia weight influences fuel economy. This can also be said of peak horsepower as a factor, because if it is high, fuel economy is low, whereas a low peak horsepower means good fuel economy. Blending the two factors together in the horsepower-to-inertia-weight ratio, however, will eliminate all recognizable tendencies.

The reason why engines of relatively low performance consume less fuel is that when they are tested moving a certain inertia weight through the fuel economy cycles, less powerful engines can be run closer to their points of minimum fuel consumption. Furthermore, less powerful engines are smaller as a rule, which makes for relatively low friction and pumping losses. On the other hand, low-performance engines need relatively high speeds to produce a given power output, but this is a factor which is more than compensated by the other influences named above.

Drivetrain

The results of a number of spot checks, together with a number of data already available, enabled us to assess the extent to which transmission and drivetrain influence fuel economy. We found that fuel economy can be improved by a maximum of 10% by changing the transmission ratios in a suitable manner.

The general tendency is for the fuel economy of an engine running at a given velocity to improve as the engine speed required to produce this velocity is reduced. However, improvements in this direction are limited by the vehicle having to retain a certain acceleration performance, which also decreases with the engine speed related to a given vehicle velocity.

The extent to which automatic transmissions influence fuel economy depends largely on the kind of cycle used in the test. In the Urban Driving Cycle, manual transmissions hardly produce any improvement whatever, whereas in the Highway Driving Cycle their introduction results in fuel economy improvements ranging between 10 and 15 %. Accordingly, the Composite fuel economy is improved by around 6 %.

Air Drag

The air drag of a vehicle is a factor which comprises both the cross-sectional area of the vehicle and its aerodynamic air drag coefficient. The cross-sectional area of a vehicle is directly related to its roominess, which is why it can be changed only within very narrow limits. On the other hand, the air drag coefficient allows manufacturers to exercise considerable discretion in design. The air drag coefficients of contemporary European vehicles range from 0.37 to 0.52, the average being 0.46.

Studies made by VW showed that reducing the air drag coefficient from 0.5 to 0.3 will improve the UDC fuel economy by 6 %, the HDC fuel economy by 22 %, and the Composite fuel economy by 11 %. It is obvious, therefore, that the influence of the air drag coefficient on fuel economy is considerable.

As a rule, however, changes of so drastic a nature cannot be made for reasons of styling. Consequently, VW have developed a trade-off method to optimize the air drag coefficient by which certain critical zones in the body of a vehicle are improved aerodynamically one after the other without changing the overall styling concept. In this way, the air drag of contemporary vehicles can be reduced by as much as 10 to 15 %, which would mean an improvement in their Combined fuel economy of between 3.5 and 5 %.

Auxiliaries

All energy-consuming auxiliaries in a vehicle are either indispensable for the operation of the engine, or help to enhance the safety and comfort of the vehicle, or both, like, for instance, the alternator.

In the engine/vehicle systems investigated by us, we found that during fuel economy testing the auxiliaries would consume power at the following rates:

Oil pump	:	1 hp approx.
Water pump	:	0.5 hp approx.
Fan	:	0.1 hp approx.
Alternator	:	Approximately 140 % of the maximum power consumption of all consumer units.
Secondary air pump	:	0.5 hp approx.
Power steering pump	:	0.5 hp approx.
Heating blower	:	0.1 hp approx.
Heater	:	0 - 0.5 l/h.
Air conditioner	:	1 hp approx.

Technical improvements in most of these auxiliaries will result in a certain amount of energy being saved; unfortunately, this would entail increases in the sticker prices.

That the alternator consumes so much energy is due to its comparatively low efficiency. Given the currents and speeds normally prevalent in fuel economy driving cycles, its efficiency is equivalent to 0.5 approx.

The heater itself does not consume any energy, provided it uses waste heat from the engine, which is the rule. Only if this waste heat should not suffice for the purpose, which is the case in extremely cold zones, additional heaters will be used which burn fuel directly.

1.3.2 Regulated Exhaust Emissions

The major emission standards applied by us were 0.41/3.4/1.0 and 0.41/3.4/0.4 gpm HC/CO/NOx.

It is a fact that the results of emission tests performed on research prototypes, on EPA certification test vehicles, and on vehicles spot-checked by EPA in the field may deviate widely. Therefore, we were faced with the task of setting engineering goals for the emission standards of 0.41/3.4/1.0 and 0.41/3.4/0.4 gpm HC/CO/NOx. We had to develop then concepts which would meet these engineering goals in the development stage assuming that they would later on meet their standards at any time, no matter when and where they would be tested.

Provided that certain conditions are met, which would mainly concern engineering, measuring and testing, production, maintenance, and EPA, we found that the HC and CO emissions would have to be at least half as low as the standard, and that the NOx emissions in the development stage should not be higher than 25 % of the applicable standard, so as to ensure that the standards would be met at all times later on.

Thus our engineering goals were 0.2/1.7/0.25 and 0.2/1.7/0.1 respectively.

The following factors are essentially the causes of scatter in emission tests:

- Driver
- Dynamometer
- Sampling equipment
- Engine
- Drivetrain
- Production tolerances.

In all field tests, the maintenance condition of the vehicles must be taken into consideration as well.

In addition, we have to take into account that the dynamometers, equipment, and drivers used for development testing and for Field Compliance Testing in all probability will not be the same.

Empirical data on the effects of total vehicle mileage are available from vehicle types already in the field. In addition, EPA Durability Test results are also available. There are significant differences between these two sets of data because of the different vehicle loads and different aging conditions in customers use.

But there are no empirical data on field and EPA Durability for vehicles in the research stage, i.e. Modification Codes 13 - 20.

Taking all this into consideration, it is difficult to arrive at any projection regarding the field behaviour of an engine/vehicle system still in the research stage.

For this reason we set demands for low emission engineering goals. They were indeed tough, but there is a chance that they may be reached after a sufficiently extended period of development, provided everyone concerned is prepared to collaborate. The first objective is to ensure that there is no deterioration in the engine and all other parts which control emissions, disregarding for the time being the oxygen sensor and the 3-way catalyst. In other words: The deterioration factor of the engine must not exceed 1. As far as the emissions of HC and CO are concerned the efficiency of the oxygen sensor and 3-way catalyst must also remain constant over 50,000 miles in the field. At the moment, it seems absolutely unrealistic to postulate that the NOx efficiency of oxygen sensors and 3-way catalysts should be equally constant. For this reason, we are allowing a deterioration factor of 2 in NOx emissions.

Assuming that the emissions of a vehicle in the development stage have been successfully reduced to less than half the permissible standards, there is another requirement which has to be met; namely, that the performance of all engine parts on which emissions depend, as well as the performance of the emission test procedure and the standards of mass production are so reliably uniform that the emission of any vehicle will remain below the standards, no matter when or where it is tested.

Thus we established our engineering goals from the emission standards by dividing the HC and CO limits by 2, and by dividing the NOx limit by 4.

We had to meet the following standards:

1) Uncontrolled		
2) 1.5/15/3.1	gpm HC/CO/NOx	(U.S. Federal 1976)
3) 0.9/9/2.0	"	(California 1976)
4) 0.41/3.4/1.0	"	(U.S. Federal 1981)
5) 0.41/3.4/0.4	"	(Research)

In addition to 0.41/3.4/0.4, levels 1.5/15/3.1 and 0.9/9/2.0 were the only regulated exhaust emission standards at the time of concluding this Contract.

The current standards we were able to meet with the necessary margin of safety by applying conventional concepts, such as carburetors and catalysts as well as on-off EGR in some cases. Current standards are relatively close to current production conditions, which is why the respective engineering goals, relatively speaking, are not as strict as those of the advanced standards, although in absolute figures they can be much stricter because the standards themselves are much higher. Thus, for example, the engineering goal related to the '81 CO emission standard is 1.7 gpm. Although this is equivalent to half the standard it is only 1.7 gpm below the standard. If, by way of comparison, you look at the results related to the '76 Federal standard you will find that all of them are much farther below that standard than 1.7 gpm.

In the course of the development work required to meet the engineering goals of the advanced standards we found that

- the standards of '81 require the use of closed loops in each case;
- in addition to this, in the 3,000 lbs inertia weight class proportional EGR is required in each case to meet the standards of '81, whereas there is no demand for EGR in the IW class of 2,250 lbs;
- with the sole exception of the 1.3 liter engine, all other engines have to have clean-up catalysts with secondary air injection to meet the '81 standards;
- the 1.3 liter engine is capable of meeting the research standard engineering goal of 0.1 gpm NOx as well without clean-up catalyst, secondary air injection, and EGR.

The HC and CO emissions of our concepts always kept below the engineering goals of the advanced standards. We even met in all cases the engineering goal of 0.25 gpm NOx, which is related to the '81 standard, but we were not equally successful as regards the goal of 0.1 gpm, which belongs to the research standard of 0.4 gpm NOx. The 1.3 liter engine did just about meet this goal, because in this engine pumping and friction losses are at a minimum. The 1.6 liter engine was quite problematic already (0.12 - 0.13 gpm NOx), and it was even more difficult to get somewhere with the 5-cylinder engine (0.15 gpm NOx).

As far as the vehicle system is concerned, the only important factor influencing emissions in a significant way is that of inertia weight. This can be demonstrated by comparing Modifications 01 and 02, 05 and 06, 09 and 10, 13 and 14, as well as 17 and 18. The transition from 2,250 to 3,000 lbs inertia weight entails emission increases which, in the case of hydrocarbons, amount to an average of 20 %, in the case of carbon monoxide to 50 %, and in the case of nitrogen oxides to 26 %, which latter figure may drop to 0 % if EGR is used only in the heavier vehicle of the two.

1.3.3 Startability and Driveability

All startability and driveability tests began with cold start at an overall vehicle temperature of either -10°, +10°, or +30 °C. Tests were conducted both at sea level and at an elevation of 1,600 m.

Figure 5.4 shows the ratings of start-up performance, indicating that the best startability is obtained from K-Jetronic engines.

Following the cold start, the vehicles were run through a VW driveability cycle. This cycle consists of a brief idle phase preceding a number of different acceleration phases. The vehicle acceleration is recorded. At the end of the test, the driver evaluates each phase. All phases are then combined to engine temperature ranges, i.e., start-up phase, cold idle, acceleration phases under start-up conditions, first warm-up, second warm-up and hot-engine driveability. The results obtained at the various temperature levels are accumulated and weighted. The resulting subtotals are added up to a grand total representing the driveability of the vehicle.

Figures 1.3.4 and 1.3.5 show the cumulative startability and driveability of the various Modification Codes and indicate that vehicles with fuel injection have better driveability ratings than vehicles with carburetors. However, it should not be concluded that lower emission levels are associated with better driveability. The improved driveability in the subject case is a result of the combination of fuel injection (K-Jetronic) and closed-loop system for compliance with stringent emission standards.

All engine/vehicle systems investigated by us met the current VW startability and driveability standards when tested at -10, +10, and +30 °C at sea level and at +10 °C at an elevation of 1,600 m.

As a first approximation, we may say that any improvement of startability and driveability entails a loss of fuel economy.

1.3.4 Performance

For all engine modifications concerned in this study, engine maps were drawn up from dynamometer tests. The data contained in these maps were used in conjunction with data concerning the drivetrains and vehicles to compute the passing performance and gradeability of all engine/vehicle systems. The results of these computations are listed in Table 1.3.3.

There are significant findings regarding the influence of inertia weight, air drag, transmission ratio, and maximum engine torque over engine speed, but it is impossible to discern any influence of the emission levels, not even an indirect one, because the factors just named are independent of the emission level. In other words, it can be seen that low-emission K-Jetronic engines are quite able to keep up with other engines.

As is shown in Fig. 1.3.3 an engine performance increase always entails fuel economy losses.

1.3.5 Noise

Our studies of the noise emitted by the various vehicles enable us to make four statements:

- Introducing more stringent exhaust emission regulations does not necessarily entail an increase in the emission of noise, although the total number of noise-generating devices is increased. This may be due to the introduction of silencing devices, such as catalysts.
- The higher the engine displacement, the lower the noise emission will be, because bigger engines run at lower speeds.
- The higher the inertia weight of a vehicle, the lower will be the noise emission, because heavier vehicles can be soundproofed more effectively.
- Generally, the bigger the engine and the vehicle the lower the noise emission, which is why the demands for low noise and good fuel economy may be said to be counterproductive.

The amount of data available now only enables us to make these qualitative statements. It is not sufficient for any quantitative conclusions.

1.3.6 Unregulated Exhaust Emissions

By means of spot checks in which only the 1.6 liter Rabbit engine was involved we tried to establish the extent to which the unregulated emissions of HCN and SO_4 are influenced by emission control concepts.

In agreement with the results of studies performed earlier on we found that the emissions of HCN drop together with those of HC and CO, although the drop at the introduction of catalysts is not quite as steep as that of HC and CO.

Sulfate emissions display not as clearly identifiable trends. That the emission of sulfate shows a tendency to grow as the temperature of the 3-way catalyst increases is reflected by the fact that the sulfate emissions of Modification Code 17, which is equipped with a 3-way catalyst, show a tendency to rise from UDC via SET (Sulfate Emission Test) to HDC, i.e. parallel to an increase in engine power output and, therefore, to an increase in the average temperature of the catalyst.

Whatever the tendencies displayed by the various emission control concepts may be, it may safely be stated that the emissions of HCN and SO_4 are extraordinarily low even in the uncontrolled vehicle, being in the range of 0.01 gpm and far below.

1.3.7 Cost

All low-emission concepts and in particular those with a NOx level of 1.0 gpm are a positive engineering contribution. However, the financial aspects were to be taken into account, too. We made an analysis of this problem on the basis of sticker prices.

We expressed the cost of all emission control concepts in terms relative to the cost of comparable uncontrolled engines, with the sticker price of the uncontrolled engine being 100, i.e. we computed the percentage by which the sticker prices of the controlled concepts exceeded those of the uncontrolled engines. The following concepts are comparable in that respect:

a) 01; 05; 09; 13 and 17,

b) 02; 06; 10; 14 and 18,

c) 03; 07; 11; 15 and 19,

d) 04; 08; 12; 16 and 20 (see Table 1.3.3).

Comparison a) shows that the introduction of the emission controll concept entails a price increase of approximately 20 %. Another 60 % is added when the system is introduced to comply with the low emission levels.

This particular step has even more pronounced effects on b). The transition from the present emission levels to those of the future will entail a 70 % cost increase, largely caused by sophisticated EGR systems and secondary air pumps.

In relative terms, the cost of the small engine will increase somewhat more steeply (c)) within the range of present emission standards because of its lower standards remains about the same as with a larger engine. The cost of complying with the strict emission standards of the future is lower in absolute and relative terms for this particular engine.

As far as the larger engine is concerned (comparison d)) it should be borne in mind that the transition from Modification Codes 4; 8 and 12 to Modification Codes 16/20 means the switch from a 4-cylinder to a 5-cylinder engine. The adding-on of the additional cost to a high base price, however, results in a comparatively low relative cost increase.

The introduction of the K-Jetronic fuel injection with the closed loop system and 3-way catalyst plus clean up catalyst for compliance with 1981 Federal Standards results in an engine cost increase by 60 ÷ 70 % compared to engines for present emission levels.

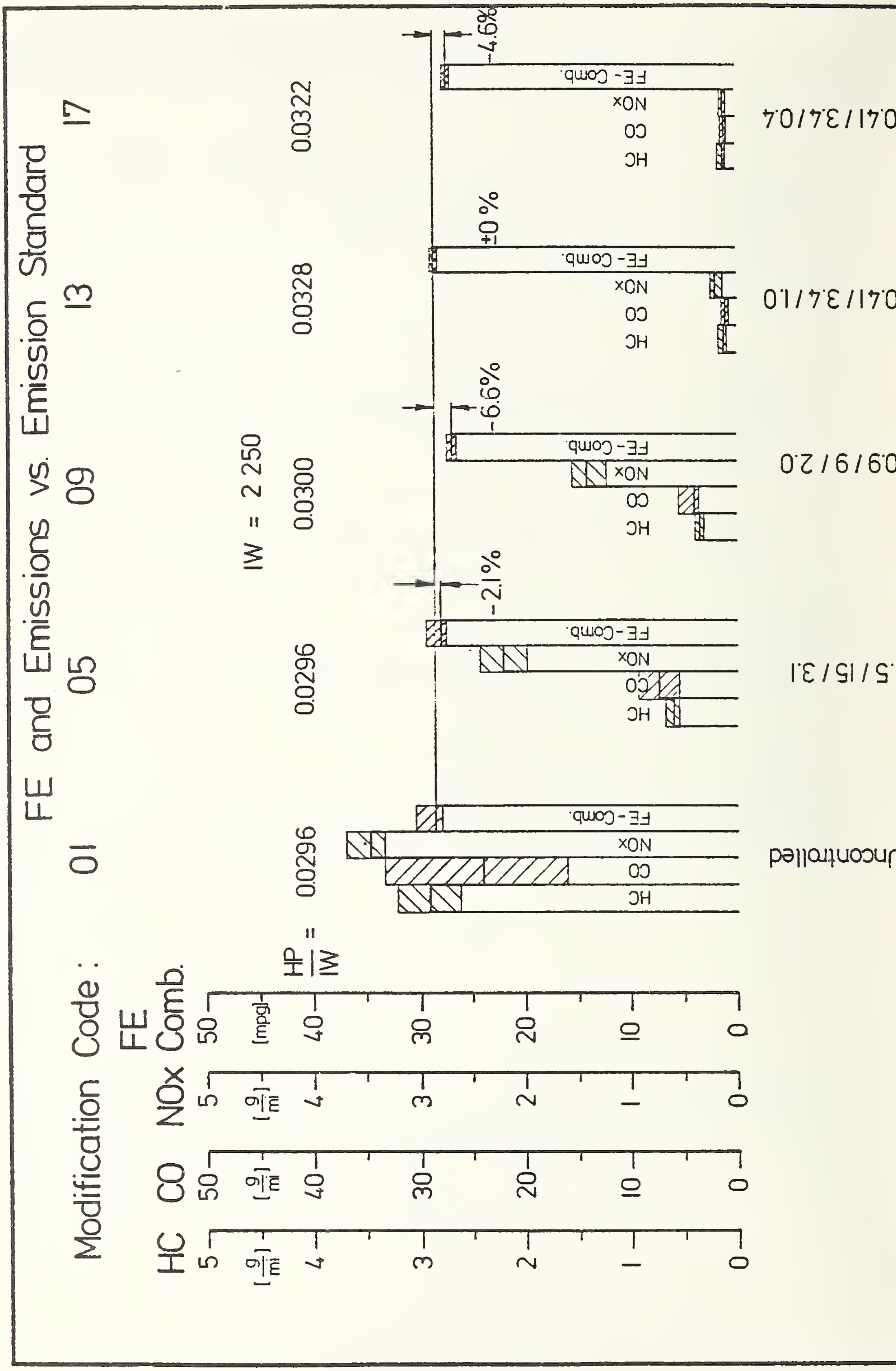

FE and Emissions vs. Emission Standard
Modification Code : 01 05 09 13 17
HP/IW = 0.0296 0.0296 0.0300 0.0328 0.0322
IW = 2 250
HC [g/mi]
CO [g/mi]
NOx [g/mi]
FE Comb. [mpg]
-2.1%
-6.6%
±0 %
-4.6%
HC
CO
NOx
FE-Comb.
Uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 1.3.1

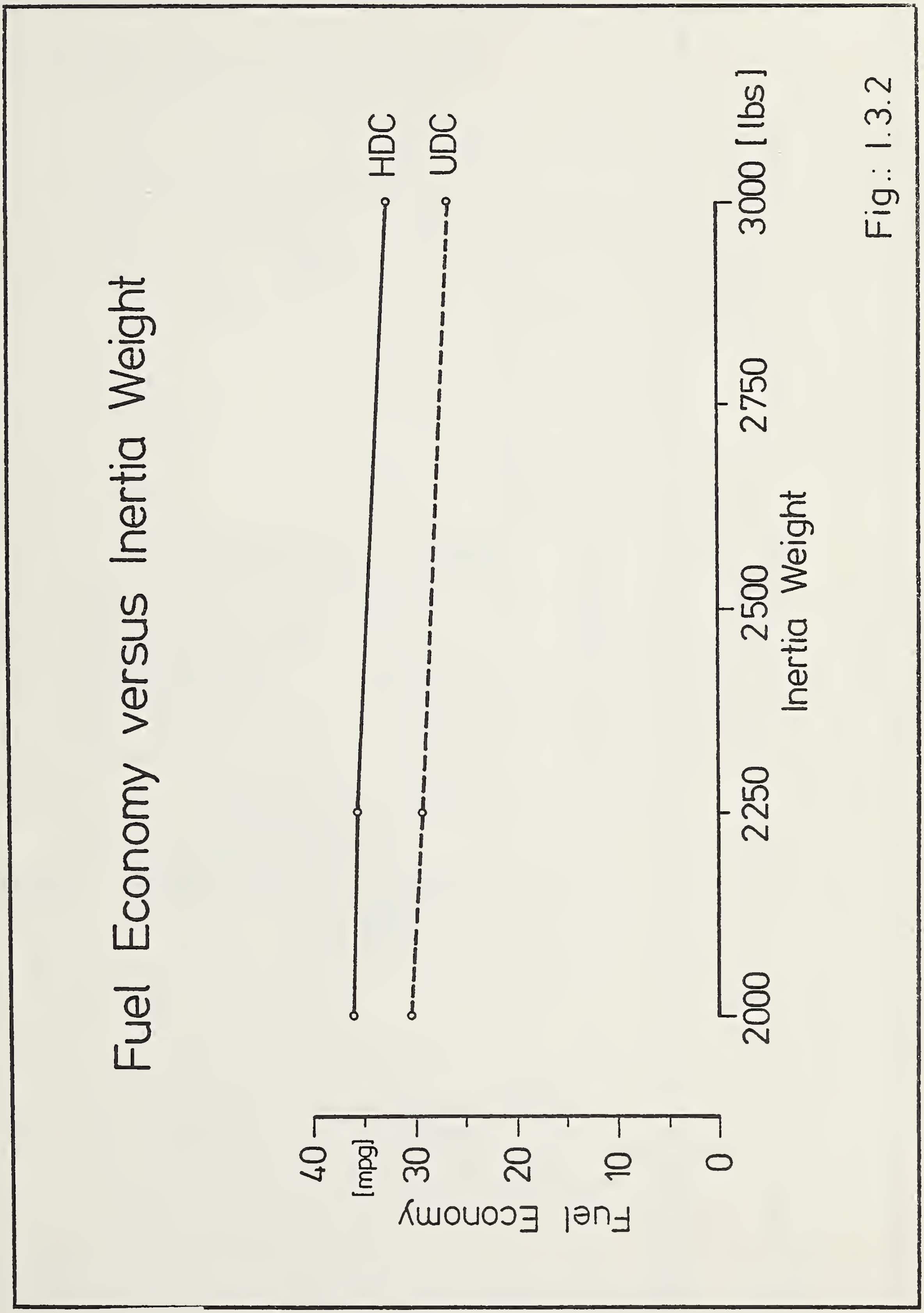

Fuel Economy versus Inertia Weight
HDC
UDC
Fuel Economy [mpg]
40
30
20
10
0
2000
2250
2500
2750
3000 [lbs]
Inertia Weight

Fig.: I.3.2

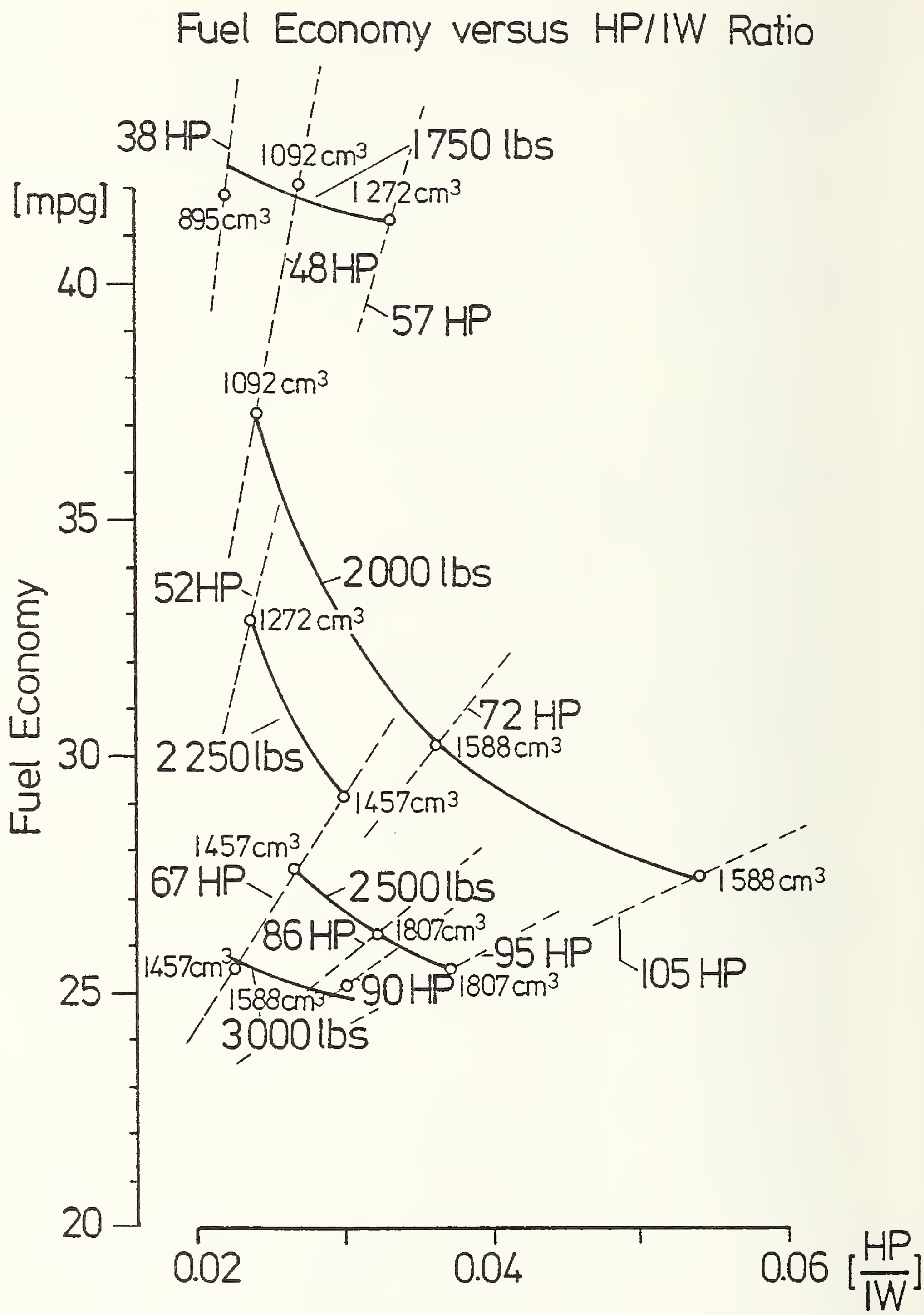

Fuel Economy versus HP/IW Ratio
[mpg]
Fuel Economy
40
35
30
25
20
0.02
0.04
0.06
[HP/IW]
38 HP
1092 cm3
1750 lbs
1272 cm3
895 cm3
48 HP
57 HP
1092 cm3
2000 lbs
52HP
1272 cm3
72 HP
1588 cm3
2 250 lbs
1457cm3
1457cm3
67 HP
2 500 lbs
1588 cm3
86 HP
1807cm3
95 HP
105 HP
1457cm3
1807 cm3
90 HP
1588cm3
3000 lbs

Fig.: 1.3.3

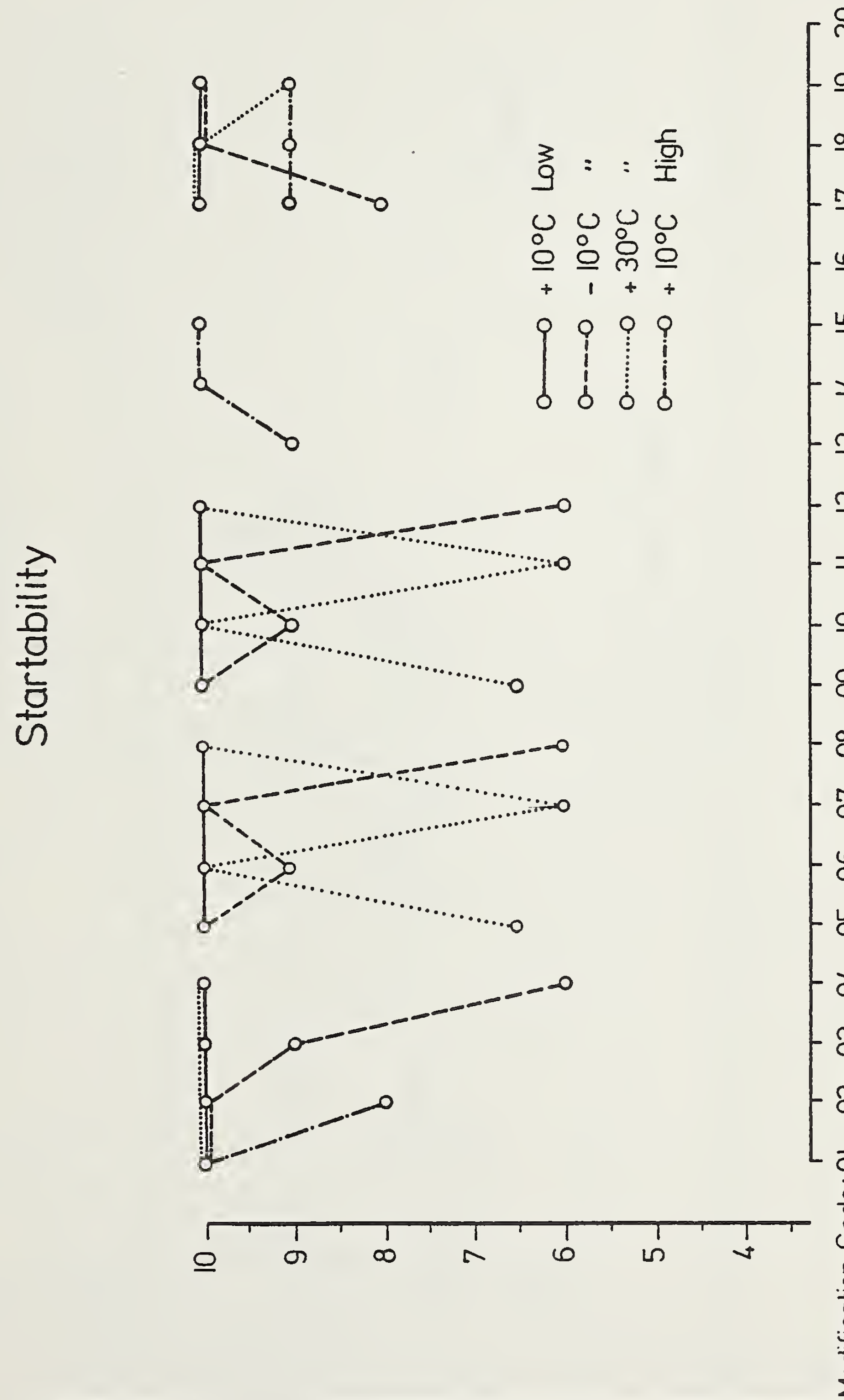
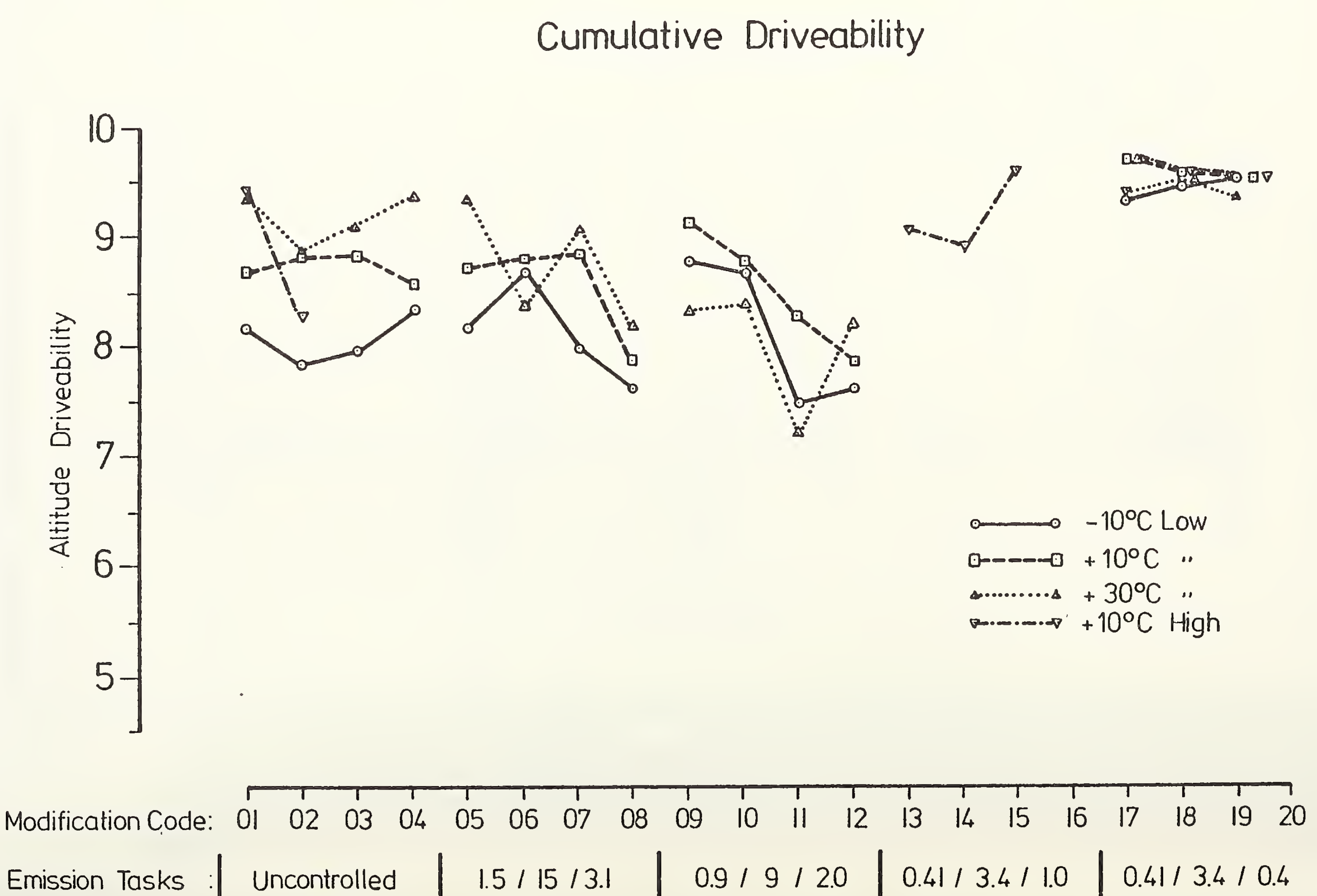

Startability
10
9
8
7
6
5
4
Modification Code: 01 02 03 04 05 06 07 08 09 10 11 12 13 14 15 16 17 18 19 20
+ 10°C Low
- 10°C "
+ 30°C "
+ 10°C High

Fig.: I.3.4



Cumulative Driveability
Altitude Driveability
10
9
8
7
6
5
-10°C Low
+10°C "
+30°C "
+10°C High
Modification Code: 01 02 03 04 05 06 07 08 09 10 11 12 13 14 15 16 17 18 19 20
Emission Tasks : Uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 1.3.5

Modification Code	Inertia Weight 2 250	Inertia Weight 3 000	F × C_D	Dyn. Wheel Diameter [cm]	Transmission Ratios Gears 1.	2.	3.	4.	Axle	No. of Cylinders	Displacement cu.in	Compression Ratio	Emission Standard [gpm] HC	CO	NOx	Engineer Goals [gpm] HC	CO	NOx	Carburetor	K-Jetronic	Decel. Control	Spark adv. diaph.	Spark ret. diaph.	Air Syst. Pump	Air Syst. Self aspir.	EGR-Contr. On-Off	EGR-Contr. Proportional	Cat. System 3"	Cat. System 6" Ox Cat.	Cat. System 6" 3-way	Closed Loop
01	×		0.77	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	uncontrolled			—	—	—	×			×									
02		×	"	"	"	"	"	"	"	"	"	"		"		—	—	—	×			×									
03	×		0.79	54.8	"	2.05	1.35	0.96	4.06	"	77.6	8.2		"		—	—	—	×			×									
04		×	0.78	60.0	"	1.94	1.29	0.91	4.44	"	96.9	"		"		—	—	—	×			×									
05	×		0.79	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	1.5	15	3.1	—	—	—	×			×	×		×			×			
06		×	"	"	"	"	"	"	"	"	"	"	"	"	"	—	—	—	×		×	×	×		×	×		×			
07	×		"	54.8	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"	"	"	—	—	—	×			×	×		×			×			
08		×	0.78	60.0	"	1.94	1.29	0.91	4.44	"	96.9	"	"	"	"	—	—	—	×		×	×	×		×	×		×			
09	×		0.79	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	0.9	9.0	2.0	—	—	—	×		×	×	×		×	×			×		
10		×	"	"	"	"	"	"	"	"	"	"	"	"	"	—	—	—	×		×	×	×		×	×			×		
11	×		"	"	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"	"	"	—	—	—	×		×	×	×		×	×			×		
12		×	0.78	60.0	"	1.94	1.29	0.91	4.44	"	96.9	"	"	"	"	—	—	—	×		×	×	×		×	×			×		
13	×		0.79	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	0.41	3.4	1.0	0.2	1.7	0.25		×	×	×	×	×					×	×	×
14		×	"	"	"	"	"	"	"	"	"	"	"	"	"	"	"	"		×	×	×	×	×			×		×	×	×
15	×		0.71	54.8	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"	"	"	"	"	"		×		×	×							×	×
16		×	0.84	60.0	3.60	2.13	1.36	0.97	4.11	5	130.8	8.0	"	"	"	"	"	"		×	×	×	×	×			×		×	×	×
17	×		0.79	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	0.41	3.4	0.4	0.2	1.7	0.1		×	×	×	×	×					×	×	×
18		×	"	"	"	"	"	"	"	"	"	"	"	"	"	"	"	"		×	×	×	×	×			×		×	×	×
19	×		0.71	54.8	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"	"	"	"	"	"		×	×	×	×							×	×
20		×	0.84	60.0	3.60	2.13	1.36	0.97	4.11	5	130.8	8.0	"	"	"	"	"	"		×	×	×	×	×			×		×	×	×

Engine / Vehicle Systems Tab.: I.3.1

Modification Code	Peak Horsepower	Max. Torque [kg×m]	UDC [mpg]	HDC [mpg]	Comb. [mpg]	HC [gpm]	CO [gpm]	NOx [gpm]	SO4 [mg/mi] UDC	SO4 [mg/mi] SET	SO4 [mg/mi] HDC	HCN [mg/mi] UDC	HCN [mg/mi] SET	HCN [mg/mi] HDC	Noise DB [A] Idle ex.*	Noise DB [A] Idle in.**	Noise DB [A] Cruise 30 mph ex.	Noise DB [A] Cruise 30 mph in.	Noise DB [A] Acceleration from 30 mph ex.	Noise DB [A] Acceleration from 30 mph in.
01	66.7	11.5	24.8	35.1	28.5	2.91	23.8	3.44	0.94	1.08	1.21	10.1	8.9	6.48	70.0	56.0	62.5	63.0	73.5	74.5
02	"	"	22.5	29.8	25.3	3.10	29.3	4.34												
03	60.4	9.4	26.9	40.9	31.8	2.44	26.5	2.02							71.0	56.0	62.0	65.5	74.0	81.5
04	84.9	12.6	21.1	31.3	24.8	3.13	27.9	3.74							71.0	49.0	60.5	61.0	72.5	73.5
05	66.7	11.5	23.9	35.1	27.9	0.59	7.26	2.18	0.62	0.55	0.45	12.5	6.2	4.4	76.0	59.0	62.5	64.0	72.0	75.5
06	67.6	11.9	21.7	30.5	25.0	0.71	9.69	2.00												
07	60.4	9.4	24.4	34.0	27.9	0.95	8.08	2.32							72.0	56.0	62.0	65.5	73.0	78.5
08	84.3	12.3	21.9	31.6	25.6	1.26	10.6	1.77							71.5	51.0	61.5	61.1	73.0	74.3
09	67.6	11.9	23.3	32.3	26.6	0.34	4.03	1.40	0.54	0.50	0.55	5.4	2.7	2.2	77.5	57.0	62.0	64.5	72.0	71.0
10	"	"	22.0	30.3	25.1	0.40	6.50	1.76												
11	55.8	9.0	24.6	35.5	28.3	0.72	6.67	1.52							71.5	55.5	61.5	65.0	72.0	78.5
12	84.3	12.3	20.9	30.8	24.4	0.62	7.63	1.74							72.0	50.5	61.5	59.5	73.5	75.0
13	73.8	11.2	25.3	33.7	28.5	0.12	0.81	0.18							69.0	54.0	61.0	63.5	72.5	78.0
14	"	"	23.5	30.6	26.2	0.16	1.24	0.18												
15	58.9	9.1	26.6	34.8	29.7	0.15	1.52	0.13							74.0	54.0	61.0	63.0	71.0	73.0
16	101.7	16.0	16.3	24.2	19.1	0.10	1.30	0.18							70.5	49.0	60.5	58.0	71.5	73.0
17	72.5	11.3	24.0	32.4	27.2	0.13	0.87	0.12	0.58	0.79	0.94	3.34	0.66	0.82						
18	74.1	10.9	20.5	29.1	23.6	0.16	1.51	0.13												
19	58.8	8.9	24.6	36.5	28.8	0.23	1.66	0.10												
20	102.6	16.0	15.5	23.0	18.2	0.06	1.28	0.15												

* exterior
** interior

Performance, Fuel Economy and Emissions

Tab.: I.3.2

Modification Code	Startability* Low Altitude -10°C	Startability* Low Altitude +10°C	Startability* Low Altitude +30°C	Startability* High +10°C	Driveability* Low Altitude -10°C	Driveability* Low Altitude +10°C	Driveability* Low Altitude +30°C	Driveability* High +10°C	Passing Ability** (FR, Vol. 34, No. 99 5-23-1969) Low-speed Distance [mi]	Low-speed Time [sec]	High-speed Distance [mi]	High-speed Time [sec]	Gradeability [%] 1.	2.	3.	4.	Weight [kg]	Width [cm]	Length [cm]	Height [cm]	Cost
01	10.0	10.0	10.0	10.0	8.13	8.65	9.33	9.37	0.071	7.8	0.217	12.2	47.5	25.8	15.8	10.8	113	60	54	70	100
02	100	100	10.0	8 0	7.82	8.81	8.85	8.24	0.073	8.2	0.248	14.3	35.4	19.0	11.5	7.9	113	60	54	70	100
03	9.0	10.0	10.0		7.92	8.79	9.06		0.071	7.8	0.232	13.2	46.0	26.3	15.9	9.7	99	67	61	63	100
04	6.0	10.0	10.0		8.31	8.57	9.34		0.073	8.2	0.223	12.6	37.8	20.1	12.8	7.3	113	60	54	70	100
05	10.0	10.0	6.5		8.13	8.65	9.30		0.071	7.8	0.217	12.2	47.5	25.8	15.8	10.8	115	60	54	70	118
06	9.0	10.0	10.0		8.65	8.78	8.38		0.073	8.2	0.248	14.4	36.7	19.8	12.1	8.1	115	60	54	70	121
07	10.0	10.0	6.0		7.92	8.79	9.06		0.071	7.8	0.232	13.2	46.0	26.3	15.9	9.7	101	67	61	63	120
08	6.0	10.0	10.0		7.59	7.85	8.21		0.072	8.0	0.226	12.8	36.7	19.4	12.6	7.4	115	60	54	70	119
09	10.0	10.0	6.5		8.73	9.11	8.30		0.070	7.65	0.223	12.5	49.2	26.8	16.5	11.1	116	60	54	70	121
10	9.0	10.0	10.0		8.65	5.78	8.38		0.073	8.2	0.248	14.4	36.7	19.8	12.1	8.1	116	60	54	70	121
11	10.0	10.0	6.0		7.43	8.23	7.17		0.072	8.1	0.242	13.8	43.9	25.3	15.6	10.0	102	67	61	63	124
12	6.0	10.0	10.0		7.59	7.85	8.21		0.072	8.0	0.226	12.8	36.7	19.4	12.6	7.4	116	60	54	70	119
13				9.0				9.03	0.071	7.9	0.217	12.0	46.3	24.9	15.1	10.1	129	54	54	70	178
14				10.0				8.83	0.073	8.3	0.242	14.0	34.2	18.4	11.2	7.5	129	54	54	70	189
15				10.0				9.56	0.072	8.1	0.236	13.5	43.6	24.9	15.3	9.7	111	55	61	63	164
16									0.071	7.8	0.204	11.2	50.1	28.5	16.7	10.8	187	50	72	83	177
17	8.0	10.0	10.0	9.0	9.31	9.67	9.69	9.31	0.070	7.8	0.217	12.0	46.6	25.1	15.1	10.1	129	54	54	70	178
18	10.0	10.0	10.0	9.0	9.43	9.52	9.49	9.51	0.073	8.2	0.242	13.8	33.6	18.0	10.9	7.3	129	54	54	70	189
19	10.0	10.0	9.0	9.0	9.51	9.51	9.32	9.40	0.073	8.1	0.236	13.4	42.4	24.3	14.9	9.5	111	55	61	63	164
20									0.070	7.8	0.204	11.2	50.4	28.7	16.8	11.0	188	50	72	83	177

* Best Value 10, Sufficient 6

** Time and Distance to Pass a Vehicle 55 ft Long Traveling 20 mph (Low-speed) or 50 mph (High-speed)

Consumer Attributes

Tab.: 1.3.3

2. INTRODUCTION

This Report represents Volume II, the main body, of a three-volume report. The objective of the study reported on herein was to obtain a data base on passenger car spark ignition engines. The power range of the engines studied was from 56 to 102 horsepower and the curb weight ranged from 2250 to 3000 pounds with peak-horsepower-to-inertia-ratios between 0.022 and 0.034.

In all these engine/vehicle systems, we investigated the way in which fuel economy, unregulated emissions, and consumer attributes were affected by the introduction of certain technologies which were required to meet given emission standards of HC/CO/NOx. The most important standards were 0.41/3.4/1.0 and 0.41/3.4/0.4 gpm HC/CO/NOx respectively.

All engines were naturally aspirated. The engine/vehicle systems tested and analysed are:

a. A subcompact vehicle (VW Rabbit; 2,250 lbs inertia weight) equipped with a 4-cylinder 1,6 l inline engine (67 ÷ 74 hp).

b. Same system as a. but of 3,000 lbs inertia weight, so as to find out what the influence of inertia weight is if all other parameters are really kept constant.

c. A subcompact vehicle (VW Rabbit; 2,250 lbs inertia weight) equipped with a 4-cylinder 1,3 l inline engine (56 ÷ 60 hp). This vehicle is not introduced in the US because the customer service organisation would not accept a further engine.

d. A compact vehicle (Audi 100; 3,000 lbs inertia weight) equipped with a 4-cylinder 1,6 l inline engine (85 hp).

e. A compact vehicle (Audi 5000; 3,000 lbs inertia weight) equipped with a 5-cylinder 2,2 l inline engine (102 hp).

Our goal was to meet the following emission levels:

1) Uncontrolled

2) 1.5/15/3.1 (gpm HC/CO/NOx)

3) 0.9/9.0/2.0 "

4) 0.41/3.4/1.0 "

5) 0.41/3.4/0.4 "

Levels 2) and 3) had to be met because, in addition to level 5), these were the only emission standards regulated at the time of concluding the Contract.

According to current Federal regulations, we measured emissions and fuel economy as well, the latter both in the Urban (UDC) and in the Highway Driving Cycle (HDC) and evolved these results into a Composite fuel economy. The unregulated emissions investigated by us were sulfate and hydrogen cyanide.

The consumer attributes considered by us were startability, driveability, acceleration performance, gradability and engine system cost.

At first, we tested emission concepts in these vehicles which we thought would meet the given emission levels with sufficient safety margins. In this, we were immediately successful with catalyst/carburetor concepts in meeting both the '76 Federal Standard (1.5/15/3.1 gpm HC/CO/NOx) and the '76 California Standard (0.9/9/2.0 gpm HC/CO/NOx) because these standards were already met by our production vehicles.

To meet the '81 Federal and the research standards, we began straight away by using only fuel injection engines (K-Jetronic) equipped with 3-way catalysts.

Having defined the engineering goals applicable to those standards, we found that these goals could only be met by going through much genuine development work.

The vehicles equipped for current standards and having concluded our development work on the vehicles intended to meet the advanced standards, we tested them according to the schedule shown in Table 3. It should be noted that all emission and fuel economy measurements were repeated five times and the noise measurements with the exception of the idle noise were repeated ten times without changing the engine adjustment, so that these figures can be assumed to be statistically secure. All other figures have been obtained from single tests.

Passing performance and gradability were assessed from all pertinent engine and vehicle data by means of a computer program.

While writing this Report data available in published and unpublished literature are collected to complete information about the most important vehicle, transmission and engine impacts on fuel economy.

3. APPROACH

In this Study, the problem of fuel economy will be reviewed from the automobile manufacturer's point of view, concentration on engine aspects.

First of all, we shall briefly go into those factors which influence fuel economy and which are outside the scope of the automobile manufacturers' influence but have to be reckoned with in the manufacture of a vehicle.

Secondly, we shall deal with the possibilities of influencing the consumer and of selecting specific engine/vehicle systems in order to modify fuel economy. Following that, we will review the engines actually manufactured by VW along lines of consideration as mentioned before.

We shall then report on the engine/vehicle systems with which this study is concerned, and we shall finally describe the results of this study, making use of a number of findings published previously. Whenever a datum of this kind is used, it is identified by a number in square brackets by means of which the source of the information in question can be traced in the Bibliography appended to Chapter 7.

3.1 GENERAL

The technical quality of passenger cars is determined by the six criteria named below:

1. Fuel economy
2. Emissions (HC; CO; NOx, unregulated)
3. Safety
4. Comfort (startability, driveability, roomeness, suspension, handling, noise, vibration, wellbeing)
5. Acceleration performance
6. Durability

Generally these criteria are influenced by

a) Atmospheric conditions (weather, altitude)

b) Road conditions

c) Traffic conditions and driver behavior

d) Type of fuel

They influence and are influenced by

e) Inertia weight

f) Engine

g) Drivetrain including wheels

h) Air drag

i) Maintenance

j) Auxiliaries.

Fig. 3.1.1 shows the interdependence between the criteria given above and the parameters which influence them.

The items listed outside the contour of the Rabbit are those which cannot be influenced by the vehicle manufacturers, whereas those shown inside the contours are the items which are part of the vehicle manufacturers' responsibility.

Although a large number of parameters influencing the quality of transport cannot be influenced by vehicle manufacturers the latter still have to take all these parameters into account when designing a vehicle. Thus, for instance, they have to ensure that a vehicle may be operated safely and comfortably at both low and high temperatures, under high atmospheric pressure at sea level and under low atmospheric pressure at high altitude; always safeguarding low emissions and good fuel economy. Wellconstructed highways permitting hours of constant traveling at high speeds, gravelled or sandy roads and even field lanes should not present any problems. The same vehicle must be able to cope with the type of stop-and-go traffic encountered during rush hours, and, finally, it should satisfy the demands of a good driver and withstand the errors of a bad one.

This Study is not concerned with any safety or comfort factors unconnected with the engine of a vehicle, inertia weight being the only exception in certain respects since as a factor it is largely independent of the engine. We shall analyze it nevertheless because it is one of the factors on which fuel economy mainly depends.

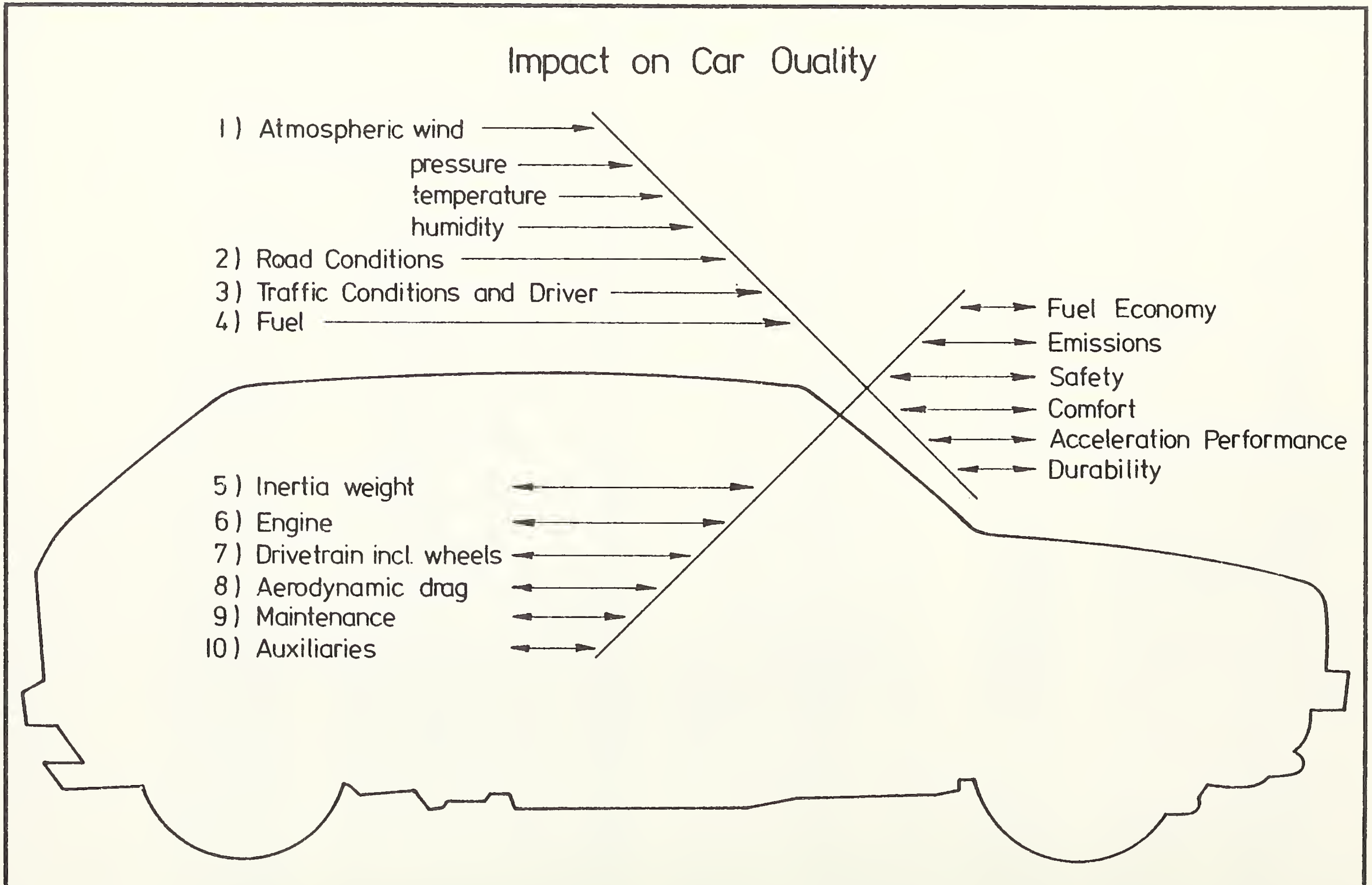

Impact on Car Quality
1) Atmospheric wind
pressure
temperature
humidity
2) Road Conditions
3) Traffic Conditions and Driver
4) Fuel
5) Inertia weight
6) Engine
7) Drivetrain incl. wheels
8) Aerodynamic drag
9) Maintenance
10) Auxiliaries
Fuel Economy
Emissions
Safety
Comfort
Acceleration Performance
Durability

Fig.: 3.1.1

3.2 PARAMETERS OUTSIDE THE MANUFACTURER'S CONTROL

3.2.1 Atmospheric Conditions

The totality of atmospheric conditions (pressure, temperature, and humidity) constitutes a boundary condition influencing the charge of cylinders. The effects of these parameters on fuel economy, emissions, and driveability are mainly due to changes in the air/fuel ratio. This may either improve or worsen operating conditions, depending on the mean conditions for which the engine is adjusted as well as on whether the environmental data are changing upwards or downwards.

Knock is also influenced by atmospheric conditions. We shall deal with this phenomen under 6.1.10.

Atmospheric Pressure

Of all atmospheric conditions the influence of pressure is most extensive, as it has a direct bearing on air density and, therefore, on air volume and the mass of oxygen contained in it, whereas the density of the fuel always remains constant. In carburetor engines, the air/fuel ratio changes with the square root of the air density.

Normally, fluctuations in atmospheric pressure caused by weather conditions will range around 5 %, going up to 8 % in extreme cases. This may cause the air/fuel ratio to fluctuate by 2.5 to 5 %. Assuming that the original ratio was stoichiometric, a reduction in atmospheric pressure will cause the fuel economy to deteriorate by approximately 3 %, whereas a corresponding increase in pressure will bring about a 1 % improvement approximately.

Atmospheric pressure will fluctuate considerably more if a car is driven at varying altitudes. The pressure difference between sea level and an altitude of 3,000 m is 30 %, which causes the air/fuel ratio to change by 14 %. Always assuming that the original ratio was stoichiometric, this means that the fuel economy will deteriorate by about 12 % when driving uphill (pressure reduction), whereas there will be a slight improvement (1 % approx.) when driving downhill (pressure increase) provided that there are no ignition failures. However, setting the air/fuel ratio of an engine at = 1 at an altitude of 3,000 m and then driving down to sea level does bring about an acute danger of ignition failure.

Probably, the mixture will then be so lean as to cause extreme HC emissions coupled with totally unsatisfactory driveability.

Engines fitted with mechanical fuel injectors must be adjustable to allow operation at varying altitudes. If this is not the case, the air/fuel ratio will change in proportion to the atmospheric pressure, which would surely render the vehicle inoperative at either high or low altitudes.

Just as fluctuations in the atmospheric pressure will affect the air/fuel ratio by way of the air density they will also influence the emission of CO, HC and NOx. A reduction in atmospheric pressure by 5 % (caused by a change in the weather) or by 30 % (caused by driving from sea level to an altitude of 3,000 m) will cause CO emissions to increase by a factor of 1.5 or 5 % compared to stoichiometric mixtures; HC emissions will increase by about 10 % or by about 80 %, whereas NOx emission will decrease to 85 % or, alternatively to 20 % of the emissions recorded in connection with stoichiometric combustion.

Compared to the emissions of stoichiometric combustion, an increase in pressure by 5 % (caused by a change in the weather) will cause CO emissions to diminish by nearly 50 % and HC emissions to diminish by approximately 5 %, whereas NOx emissions will increase by about 10 %.

The undesirable influence of atmospheric pressure can be nullified nearly entirely by installing some sort of pressure compensation equipment generally termed an 'altitude compensator'.

However, there is no way of eliminating the influence of atmospheric pressure on peak horsepower and on the engine torque at WOT. The former will increase and decrease in direct proportion to the atmospheric pressure.

Fig. 3.2.1 (1) shows fuel economy and HC, CO, and NOx as functions of atmospheric pressure as determined both by the US 75 and the US 72 warm-start tests. The data entered in the graph are averages drawn from 9 or 12 individual measurements taken at 6 or 5 different atmospheric pressure levels.

The vehicle used for these tests was a Dasher fitted with a λ -controlled 1.6 l K-Jetronic engine. The influence of atmospheric pressure may be different in a different vehicle.

Throughout the tests, relative humidity was kept constant at 50 % $\pm$ 30 %, and temperature at 20 °C $\pm$ 20 °C. The stright lines entered in the graph are regression lines. HC changes approximately by a factor of 2, and CO approximately by a factor of 4. There is a relationship between fuel economy and atmospheric pressure as well, although it is not that significant.

As measured according to the modified US 72 warm-start test procedure the influence of atmospheric pressure on emissions is similar to that found by US 75 testing, although there is a striking difference in the actual figures obtained.

Fig. 3.2.2 (1) again shows the same results, this time standardized to reflect a pressure difference of 10 mbar, which corresponds to a difference in altitude of 100 m.

Even an altitude difference as low as this one makes for a change in fuel economy of about 0.5 %.

Air Temperature

Changes of air density are inversely proportional to changes in absolute temperature, whereas air/fuel ratio changes are directly proportional to the square root of the temperature. Seasonal temperature fluctuations range around 10 % (i.e. ± 30 °C), whereas temperature differences caused by changes in altitude (e.g. from sea level to 3.000 m) range around 7 %. Because of these factors, the air/fuel ratio changes by 5 or, alternatively by 3 %. The attendant deterioration of fuel economy can reach 5 %.

Reductions in air temperature will also bring about a worsening in the cold-start and warm-up fuel economy as sinking air temperatures demand progressively richer fuel mixtures.

Via the air/fuel ratio the air temperature also influences the CO, HC and NOx emissions. Assuming that the absolute temperature increases by 10 % (i.e. the maximum seasonal deviation from the mean) the volumetric emission of CO will triple approximately compared to the emission at stoichiometric combustion. HC emissions are increased by approximately 20 %, whereas NOx emissions are reduced by 30 %. Under the same conditions, a 10 % temperature decrease will reduce CO by 70 % and HC by approximately 10 %, whereas NOx emissions will increase by about 20 %.

But the fact that the cold-start and warm-up mixture will grow richer as the temperature goes down makes for an increase in CO and HC emissions again.

Via the air density and the air/fuel ratio, the air temperature also influences the torque at WOT as well as the peak horsepower. As the air temperature increases, the subsequent torque reduction is approximately proportional to the square root of the temperature.

As the air temperature increases, startability and driveability at cold start as well as during the warm-up period will improve noticeably. To make the best possible use of this fact modern engines are designed so that the air is sucked in from the surface of the exhaust namifold during cold start and warm-up.

Air Humidity

Relative humidity normally varies between 20 and 100 %. The upper limit of humidity is determined by the prevalent saturation pressure and is rather dependent on temperature. For instance: The air/fuel ratio deviation caused by a variation between dry air and 100 % humidity is 2.3 % at a temperature of 293 °K. If the engine is adjusted to a mean humidity of 60 % the maximum possible deviation in the air/fuel ratio amounts to 1 %. Fuel economy is affected to an even lesser degree.

Compared to stoichiometric combustion the emission fluctuations caused by the changes named above are noticeable in the case of CO, insignificant as far as HC is concerned, and clearly noticeable in the case of NOx. In addition to this, increasing the water content in the air will cause the percentage of tri-atomic inert gases to increase as well, thus lowering the peak temperature of the combustion process as well as the NOx emissions. Thus, for instance, NOx emissions will drop by about 10 % after a humidity increase from 40 to 70 %. This influence of lowering the peak temperature on CO and HC emissions is insignificant.

High humidity at temperatures ranging from 270 to 285 °K especially affects driveability during the warm-up period because under those conditions the carburetor may freeze up easily. A good way to combat this effect is to have the air sucked in from around the exhaust manifold during the warm-up period.

3.2.2 Road Conditions

Road conditions are a factor influencing mainly the design of the tires and the chassis of a vehicle, especially as far as durability is concerned.

In spite of that, road conditions will noticeably influence both fuel economy and emissions. An extremely rough surface will mean increased rolling resistance, whereas an extremely smooth surface (ice) will mean increased slippage. Compared to a well-constructed road with a good grip, the two extremes named above mean increased fuel consumption and emissions.

As these influences cannot be compensated by means of technical changes in a vehicle the surfaces of the dynamometer rollers have been standardized by law.

3.2.3. Traffic Conditions and Driver

It may be safely said that the traffic or operating conditions of a vehicle in conjunction with the personality of its driver constitute the major factors influencing both fuel economy and emissions.

With some justification, automobile manufacturers are being prompted constantly to do everything in their power to keep fuel consumption as low as possible. However, they are unable to influence the traffic conditions under which their vehicles are used.

Moreover, the extent of the influence which automobile manufacturers have on the way in which their vehicles are driven is extremely limited. On the one hand, the safety of the vehicle, of its occupants, and of the other traffic has top priority. Consequently, a driver driving a vehicle must be allowed a large amount of discretion to ensure that he can evade sudden dangers.

Thus, for instance, vehicles should have good acceleration so that other vehicles may be passed as quickly as possible. However, a driver may make full use of the acceleration performance of his vehicle when there is no actual need for it, thus increasing his fuel consumption unnecessarily.

On the other hand, manufacturers have little influence on the way in which a vehicle is driven because one of the major factors determining the attractiveness of automobiles is the fact that they can be used far more individualistically than most other means of transport.

Given the instationary operating conditions under which vehicles are running today, energy is consumed mainly to accelerate the mass of the vehicle.

The mass is composed of the inertia weight and the payload. A high payload means high absolute fuel consumption. Nevertheless, for reasons of fuel economy it would be desirable to have higher payloads. The higher the payload, the lower the percentage of fuel used for moving the inertia weight of the vehicle, or, in other words, fuel economy is much improved if six persons travel not in six passenger cars but in one.

As soon as we will succeed in rendering the speed of the general traffic more uniform by specific traffic control measures, vehicle mass will be a less significant factor.

As an example we present a comparison of fuel consumptions recorded at a constant speed of 31 mph and during the European Emission Test, which proves our point in spite of the fact that the average speed of the European test is no more than 11.8 mph because of the long idle phases.

Rabbit	1.5 1/75 Hp	1.1 1/50 Hp
31 mph Cruise	5 1/100 km	4.8 1/100 km
European Test	13.8 1/100 km	11.1 1/100 km
Fuel Saved in Constant-Flow Traffic	64 %	57 %

Although cruising at 31 mph is a theorectical borderline case it serves to illustrate the fuel saving potential inherent in a more steady flow of traffic.

As a consequence of this situation, governments - quite rightly - have standardized the two parameters 'traffic conditions' and 'driver personality' so as to be able to compare and to limit the fuel consumption and the emissions of all vehicles. These government standards are the EPA Urban and Highway Driving Cycle.

The actual extent of the influence of the driver's personality is made clear by the fact that in spite of all this standardization test results differ significantly, depending on which driver runs the vehicle through the driving cycle.

We used 4 different drivers running the same standard vehicle to investigate the influence of the driver. Our standard of comparison was an automatic driver developed specifically for the purpose (1).

Our tests were run in an air-conditioned test chamber, the temperature of which was kept constant at 20 ± 2 °C, while the humidity was kept at 50 ± 3 %, and the atmospheric pressure at 1013 ± 13 mbar.

The chassis dynamometer was adjusted very precisely before the tests began.

We used this sophisticated equipment for these tests so as to keep all parameters constant, except for that of the driver. A total of 10 US 75 cold-start tests and 16 warm-start tests (US 72, oil temperature 75 ± 2 °C) was run by each of the four drivers and by the automatic drivers as well. Including the preliminary tests, the total number of tests run was approximately 180.

Reflecting the performance of all 4 drivers as well as the automatic driver, Fig. 3.2.3 (1) shows the dispersion of the HC, CO, and NOx emission results and of the fuel economy findings obtained from the US 75 and the US 72 warm-start tests.

Fig. 3.2.4 (1) shows the same results again, this time in the form of histograms with fitted standard distribution density functions.

The dispersions produced by the automatic driver in the US 75 tests are lower than the average dispersion of each of the four drivers by a factor of 4 in the case of HC, by a factor of 2 in the case of CO, by a factor of about 2 in the case of NOx, and by a factor of about 3 in the case of fuel economy.

Compared to the driver whose results were most widely dispersed, the scatter bandwidth of the results obtained from the automatic driver is narrower by a factor of up to 7 (HC, driver No. 3).

The position as regards dispersions is approximately the same as far as the US 72 warm-start tests are concerned. However, the HC and CO results obtained from the automatic driver are more widely dispersed here. This, however, is because the dispersions due to the engine itself fluctuate with time.

The results obtained from a 'good' driver in tests conducted according to the US 75 procedure, for instance, vary by $\pm$ 16 %, whereas those obtained from a 'not-so-good' driver vary by about $\pm$ 32 %.

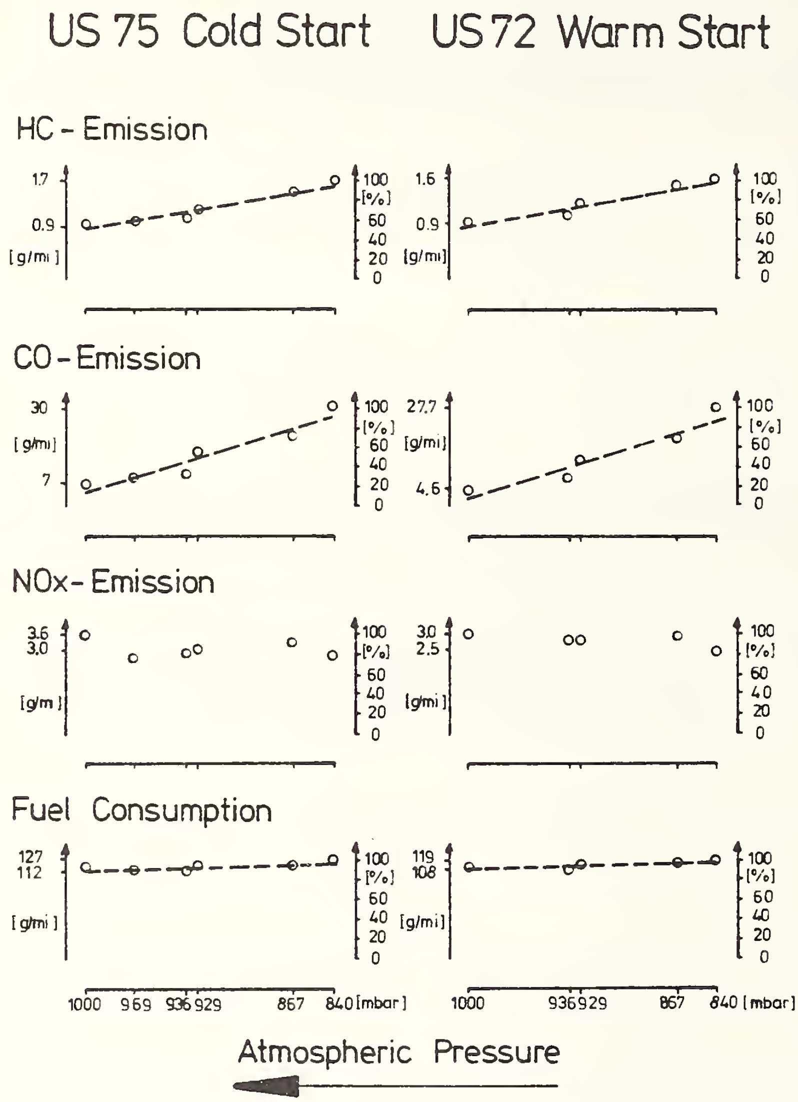

US 75 Cold Start
US72 Warm Start
HC - Emission
1.7
0.9
[g/mi]
100
[%]
60
40
20
0
1.6
0.9
[g/mi]
CO - Emission
30
[g/mi]
7
27.7
[g/mi]
4.6
NOx - Emission
3.6
3.0
[g/mi]
3.0
2.5
[g/mi]
Fuel Consumption
127
112
[g/mi]
119
108
[g/mi]
1000 969 936 929 867 840 [mbar]
1000 936 929 867 840 [mbar]
Atmospheric Pressure

Fig.: 3. 2. 1

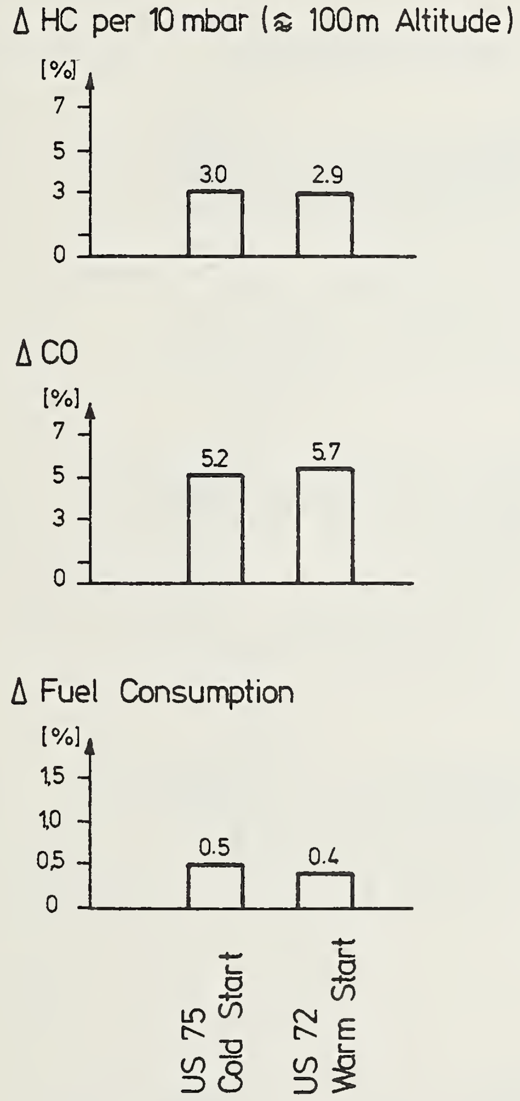

Δ HC per 10 mbar (≈ 100m Altitude)
[%]
7
5
3
0
3.0
2.9
Δ CO
[%]
7
5
3
0
5.2
5.7
Δ Fuel Consumption
[%]
1,5
1,0
0,5
0
0.5
0.4
US 75 Cold Start
US 72 Warm Start

Fig.: 3. 2. 2

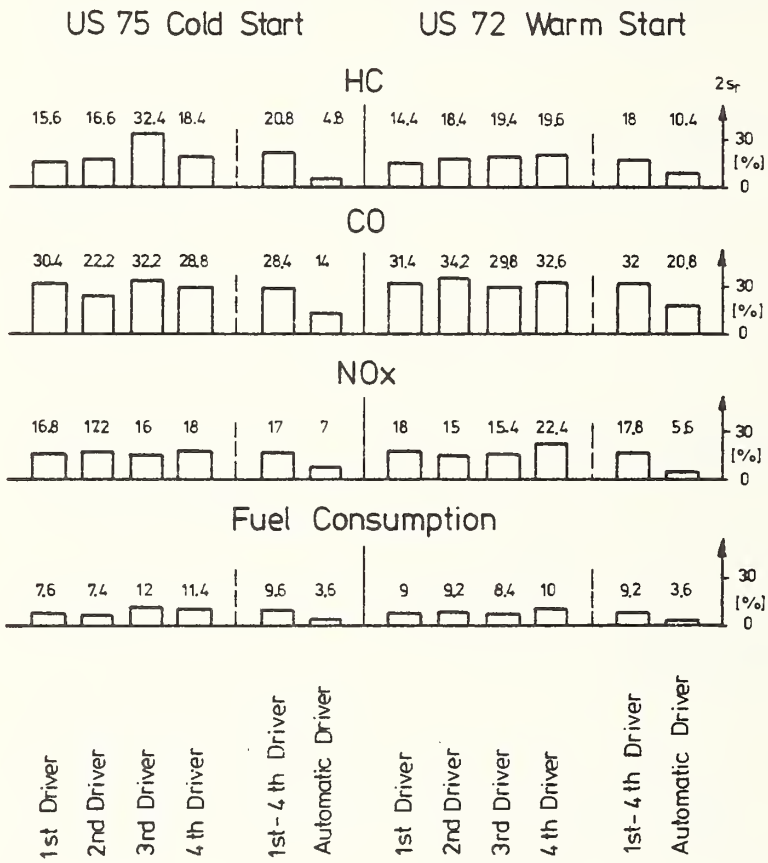

US 75 Cold Start
US 72 Warm Start
HC
2s_r
15.6 16.6 32.4 18.4 20.8 4.8 14.4 18.4 19.4 19.6 18 10.4
30
[%]
0
CO
30.4 22.2 32.2 28.8 28.4 14 31.4 34.2 29.8 32.6 32 20.8
NOx
16.8 17.2 16 18 17 7 18 15 15.4 22.4 17.8 5.6
Fuel Consumption
7.6 7.4 12 11.4 9.6 3.6 9 9.2 8.4 10 9.2 3.6
1st Driver
2nd Driver
3rd Driver
4 th Driver
1st-4 th Driver
Automatic Driver
1st Driver
2nd Driver
3rd Driver
4 th Driver
1st-4 th Driver
Automatic Driver

Fig.: 3.2.3

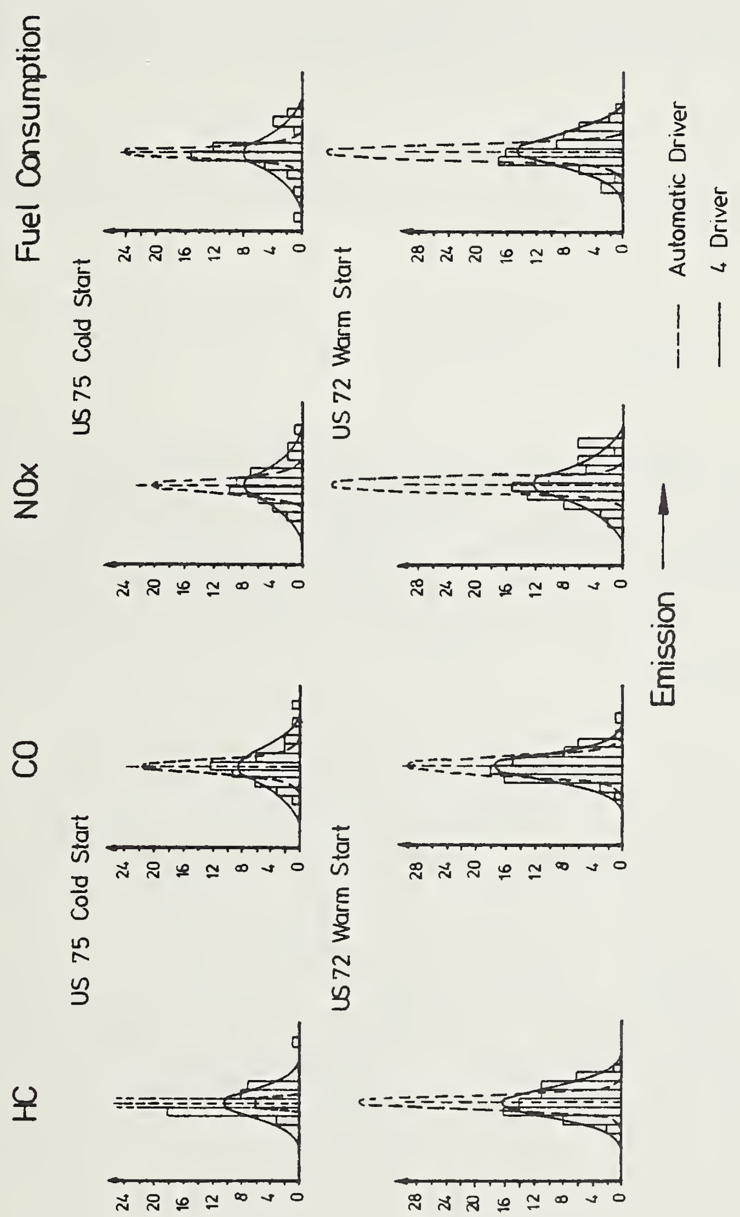

HC
CO
NOx
Fuel Consumption
US 75 Cold Start
US 72 Warm Start
Emission
Automatic Driver
4 Driver

Fig.: 3. 2. 4

3.3 PARAMETERS CONTROLLED BY AUTOMOBILE MANUFACTURERS

Basically, there are two fields of action in which automobile manufacturers can influence fuel economy, the first one being public relations and the information which accompanies the sale of a vehicle to a customer, the second one being the technical field, where automobile manufacturers are in a position to take direct and decisive action.

Among the aspects influencing fuel economy the engine is certainly foremost, and thus it is the engine which is the main subject of this investigation.

3.3.1 Consumer Information

For some years, the advertising campaigns of VW have made much of the fact that for a number of reasons VW vehicles and VW engines are among those whose fuel consumption is lowest. Moreover, VW are trying to make their customers feel that they have chosen an outstandingly economical vehicle, a Volkswagen, a symbol of economic reasoning, which is expressed not so much by the price of the vehicle itself but rather by its excellent fuel economy. This line of argument is used to ensure that customers will run their vehicles so that fuel consumption is indeed low.

Then, there has been a campaign for many years to inform customers about ways of saving fuel by careful driving. Ever since the late fifties, for instance, instruction manuals contain instructions detailing which transmission speed is the most economical to use at a given velocity. Instruction manuals today even contain separate sections stating that top speeds and maximum acceleration should be avoided, and that it is preferable to drive evenly and efficiently in order to save fuel. It is said that the economic efficiency of a vehicle mostly depends on its owners' style of driving; it is also stated that there are factors adversely affecting fuel consumption which, although not under the direct influence of a driver, can be avoided, for instance, by proper timing. Thus, very dense rush-hour traffic is mentioned as one of the factors which increase fuel consumption. Frequent driving over short distances, with the engine allowed to cool off in between, is introduced as another factor blowing up fuel consumption. Loose surfaces, sand, snow, and ice should be avoided, and driving in a line of vehicles should always be done in high gear.

On the other hand, fuel consumption is brought to an optimum level by driving over long distances at moderate speeds and without stopping overmuch. Even the highway fuel consumption can be kept low by driving below top speed. The best fuel consumption is found in the middle engine speed range, together with the best mean effective pressure.

Finally, customers are reminded to keep their vehicles in optimum condition because this is the best way for them to save fuel.

3.3.2 System Design Considerations

Fig. 3.3.1 shows the interdependence between the parameters which can be changed by the automobile manufacturers and the technical criteria, i.e. the quality of the vehicle, resulting from them, into account the comfort factors of chassis and equipment as well as safety not being taken.

We presume in this figure that all parameters will be changed so as to attain optimum fuel economy. This means reducing the inertia weight, minimizing fuel consumption of the engine using manual rather than automatic transmissions, and reducing the transmission ratio, lowering the air drag, increasing the extent of maintenance, and reducing the power reserved for auxiliary equipment.

The diagram shows the way in which all other technical vehicle criteria will change in those circumstances. In all cases, emissions improve, as does the fuel economy, or remain.

Reducing the inertia weight means a loss of comfort for many reasons, the most important one being that disturbing forces of the same magnitude affecting two vehicles of different inertia weights will not be felt to be quite as disturbing inside the heavier vehicle.

Engines offering optimum fuel economy are bound to be smaller than today's engines in order to reduce the amount of power consumed by friction inside the engine itself and in order to ensure that engines can be operated at wider average throttle aperatures than the engines of today. Of course, these two factors entail a loss of comfort. Driveability decreases as fuel economy improves, especially in cold-start and warm-up conditions, because leaner fuel-air mixtures are used than up to now.

It is obvious that automatic transmissions offer a higher degree of comfort than manual transmissions. On the other hand, higher transmission ratios are more noisy.

Air drag influences comfort because reducing the air drag entails a reduction of wind noise inside the vehicle. Intensified maintenance to ensure good fuel economy will surely increase comfort as a sort of by-product, while cutting down the amount of power allotted to auxiliaries will necessarily lead to a loss of comfort.

The acceleration performance of a vehicle increases as its inertia weight decreases. A performance increase, however, does not apply to engines offering optimum fuel economy as their stretchines is low. Moreover, their maximum power output can be expected to be low as well.

Because significant FE gains are possible only by using transmissions geared to ensure low engine speeds transmissions geared to optimum fuel economy will deteriorate acceleration performance. Reducing the air-drag also improves acceleration performance, especially at high speeds. Increased maintenance will also tend to improve acceleration. Finally, a reduction in the amount of power consumed by auxiliaries is sure to improve acceleration performance as well.

The influence of air drag, drivetrain, and auxiliaries on durability is negligible to all intents and purposes. Durability is sure to be negatively affected by a reduction in inertia weight as well as by optimizing the fuel economy of the engine, as engines of this kind generally operate under specific loads higher than those of nonoptimized engines. Here again, short maintenance intervals are likely to improve durability.

3.3.3 Engine Selection

Passenger cars may be powered by all sorts of mechanism, such as turbines, steam engines, Stirling engines, gasoline engines, Diesel engines, etc. Seen from the angle of strict fuel economy, there are only two alternatives worthy of consideration today, namely gasoline and Diesel engines, with Diesel engines enjoying a distinct advantage.

As an example, we are going to compare in the following our 1.1 l Gasoline engine (not available in the U.S.) and our 1.5 l Diesel engine, because they have the same peak horsepower. In addition, we will consider our 1.5 l gasoline engine because it is of the same size as the Diesel engine.

The specific fuel consumption of the Diesel engine is substantially lower than that of the gasoline engine, the specific fuel consumption of the gasoline engine being reached at the peak horsepower speed (see Fig. 3.3.2 and 3.3.3). Diesel engines are superior because of their favorable fuel consumption under partial load, especially under collective loads such as will occur during urban driving cycles and exhaust emission test procedures.

According to our extrapolations, Fig. 3.3.4 shows fuel economy as a function of vehicle weight based on measurements of production engines as well as of prototype and research engines which are available at present.

Subcompact cars equipped with modern Diesel engines could comply with all recommended standards. Making gasoline engines comply with recommended standards would require the installation of very expensive exhaust emission control systems.

It should be borne in mind that most production Diesel engines are not very well suited to small cars because of their unfavorable dimensions and power-to-weight ratios. Other adverse factors are high noise pollution as well as emissions of particulates and odor. But VW-Diesel engines have been developed from gasoline engines specifically for application in passenger cars. Their emissions are very low. Their suitability and functional quality are shown in Fig. 3.3.5.

While the dimensions of the Diesel engine are similar to those of the gasoline engine its weight is about 20 kg higher.

A feature of special interest is the Diesel engine's high torque over the entire speed range. According to our investigations, this means that two identical subcompact cars powered by a Diesel and a gasoline engine would be of roughly the same level of driveability, acceleration performance, and maximum speed.

To expand the scope of this comparison we extended it to include a gasoline engine which in all its components including CID is identical with the Diesel engine. This comparison is exemplified by Figs. 3.3.6 and 3.3.7 as well as by Table 3.3.1. Compared to the fuel consumption of the Diesel engine, that of a spark ignition engine of the same power is up by 26 %, and that of a spark ignition engine of identical design but a power output of 75 HP is higher even by 38 %. The partial road load curves reflect a similar situation.

In spite of their advantages we shall from now on leave Diesel engines out of consideration as the task set for this study clearly includes to gasoline engines only.

From all other gasoline engine designs, the stratified charge engine is outstanding because of its good fuel economy potential. This type of engine was developed mainly in order to reduce the exhaust emission rates, and because a special combustion process is involved, this design is a success in this respect, especially as far as nitrogen oxide emissions are concerned. As regards fuel consumption, all stratified charge engines known today do not differ from conventional spark ignition engines - there are no special advantages and no disadvantages, either. It may be that purposeful development work in this direction might open up some new vistas, but as no new developments have come up in this field so far, we shall from now on disregard the stratified charge engine.

	To Attain Good Fuel Economy	Consequences for			
		Emissions	Comfort	Acceleration Performance	Durability
Inertia Weight :	Low	Low	Low	High	Low
Engine :	Small Size	"	"	Low	"
Drivetrain :	Manual; Low Transmission Ratio	•	"	"	•
Air Drag :	Low	Low	High	High	•
Extent of Maintenance :	High	"	"	"	High
Power for Auxiliary Equipment :	Low	"	Low	"	•

Fig.: 3.3.1

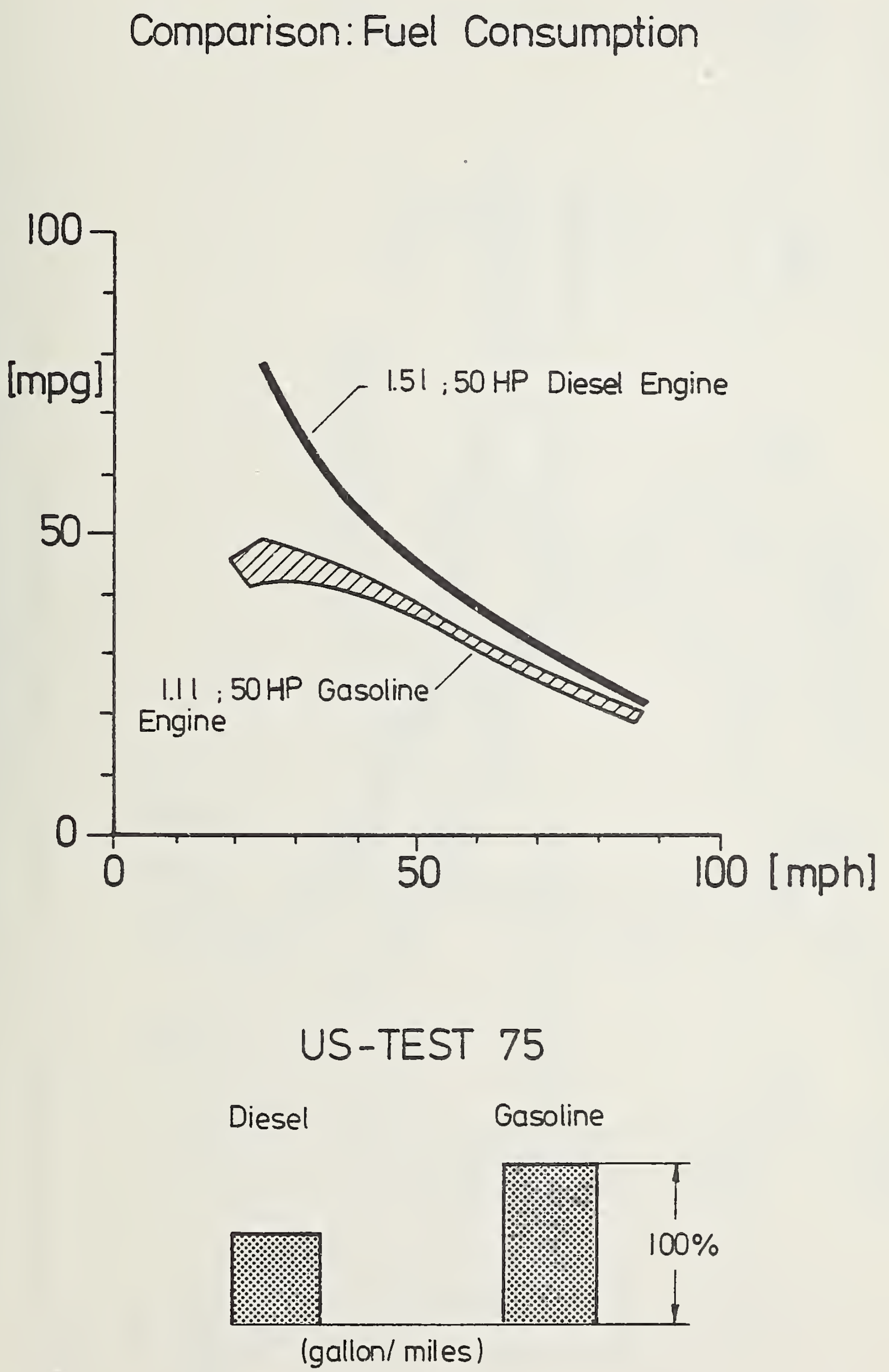

Comparison: Fuel Consumption
100
[mpg]
50
0
0
50
100 [mph]
1.5 l ; 50 HP Diesel Engine
1.1 l ; 50 HP Gasoline Engine
US-TEST 75
Diesel
Gasoline
100%
(gallon/ miles)

Fig.: 3.3.2

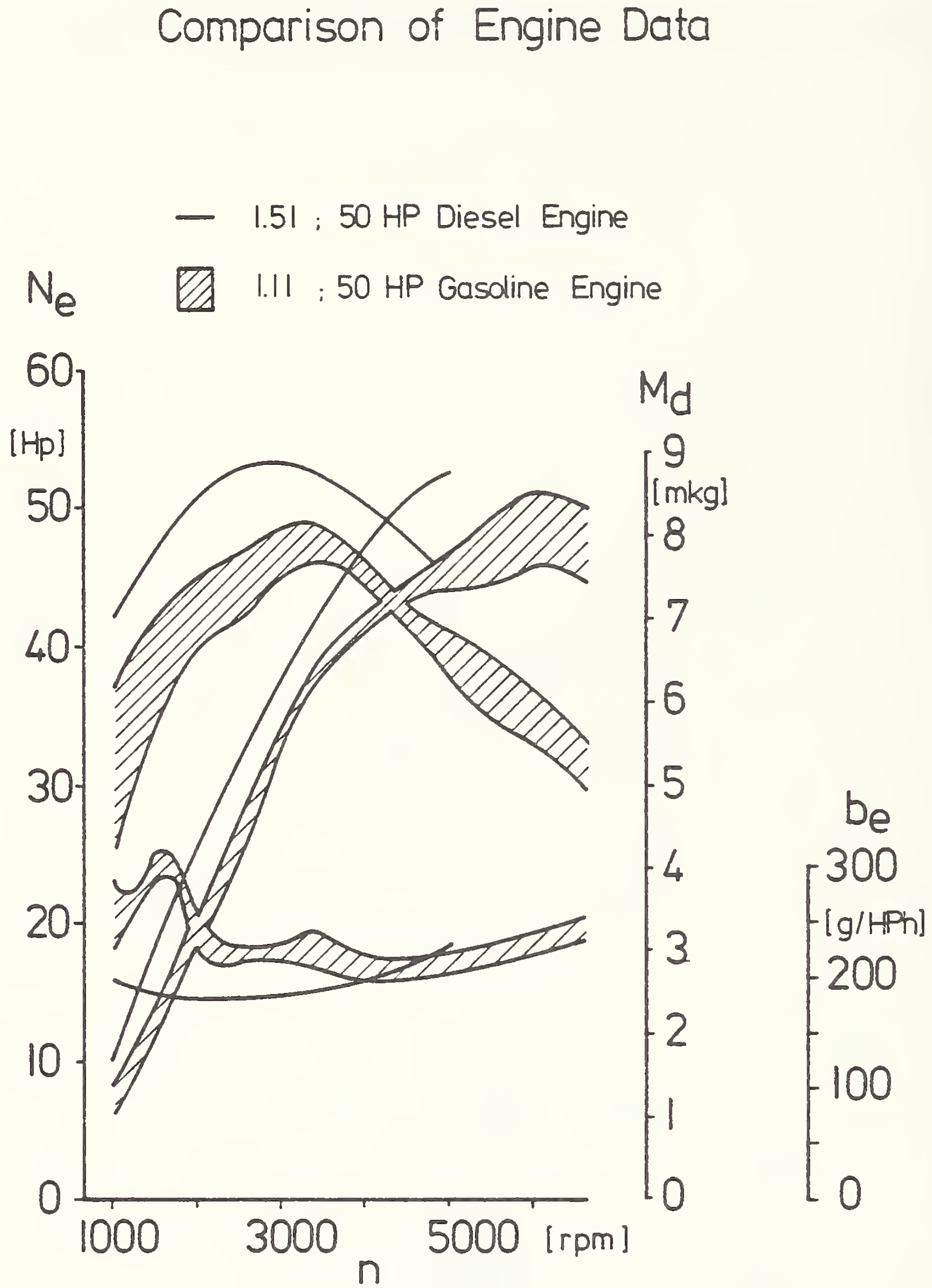
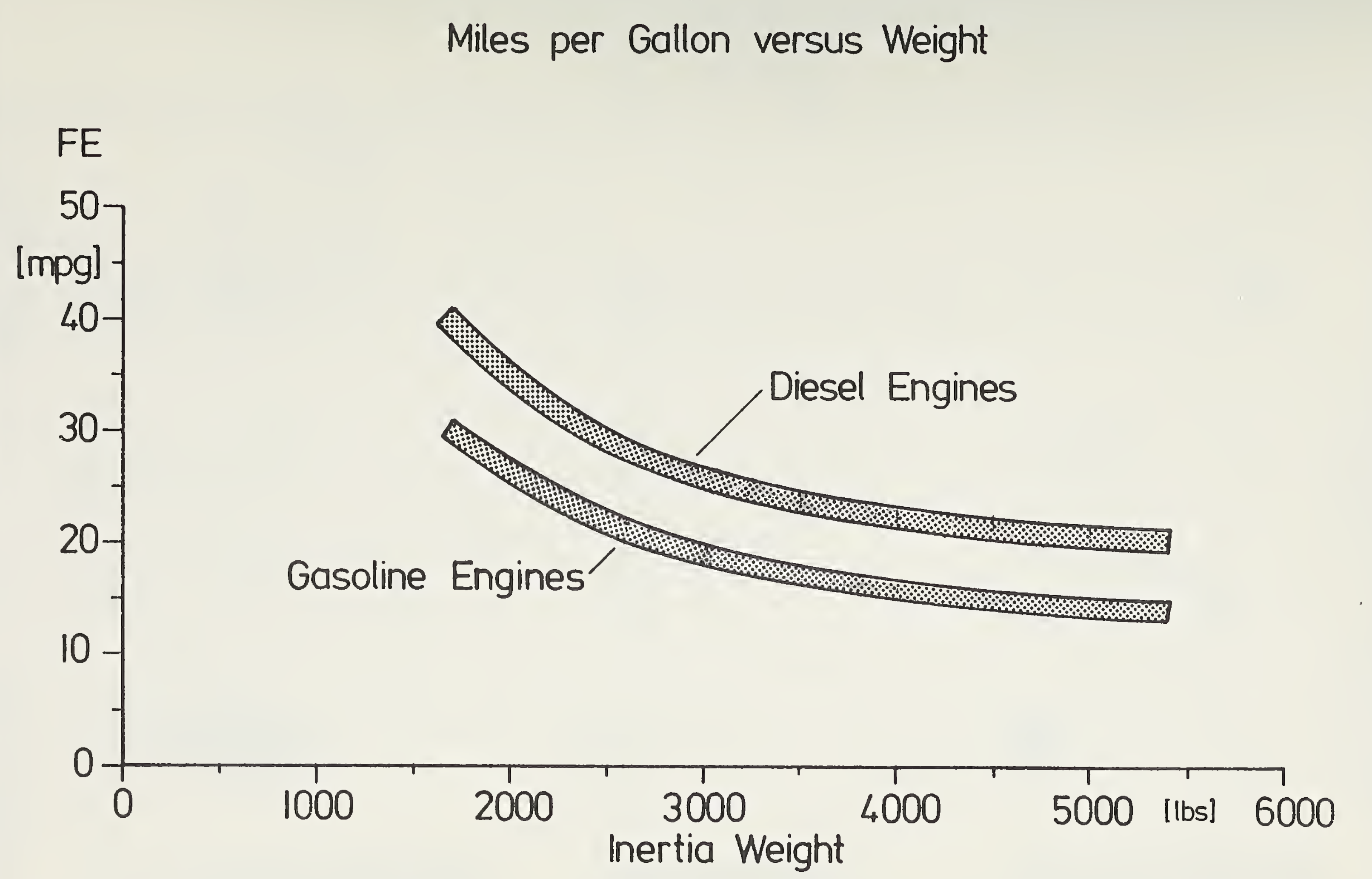

Comparison of Engine Data
1.5 l ; 50 HP Diesel Engine
1.1 l ; 50 HP Gasoline Engine
Ne
60
[Hp]
50
40
30
20
10
0
Md
9
[mkg]
8
7
6
5
4
3
2
1
0
be
300
[g/HPh]
200
100
0
1000
3000
5000
[rpm]
n

Fig.: 3.3.3



Miles per Gallon versus Weight
FE
50
[mpg]
40
30
20
10
0
Diesel Engines
Gasoline Engines
0
1000
2000
3000
4000
5000
[lbs]
6000
Inertia Weight

Fig.: 3.3.4

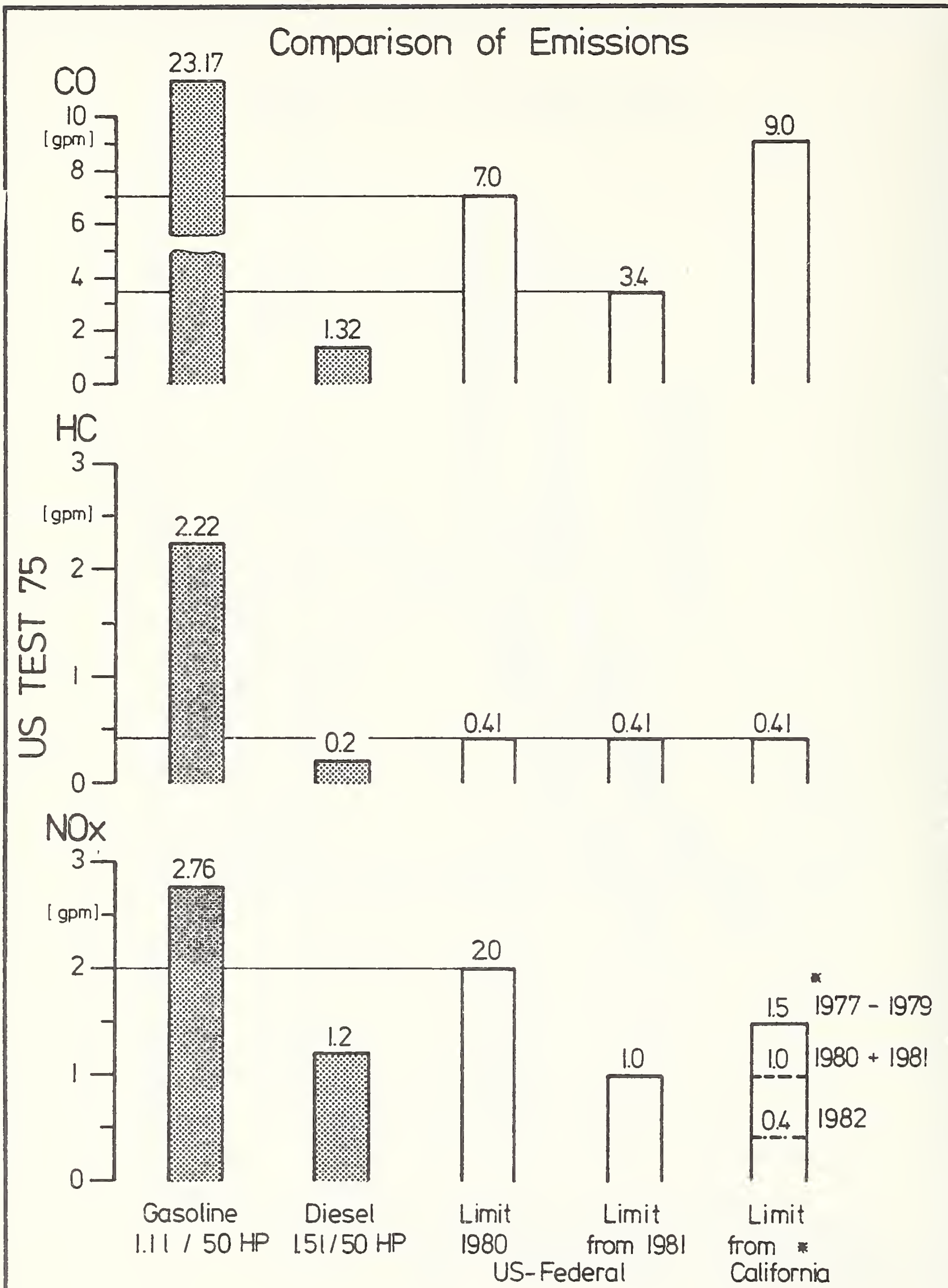
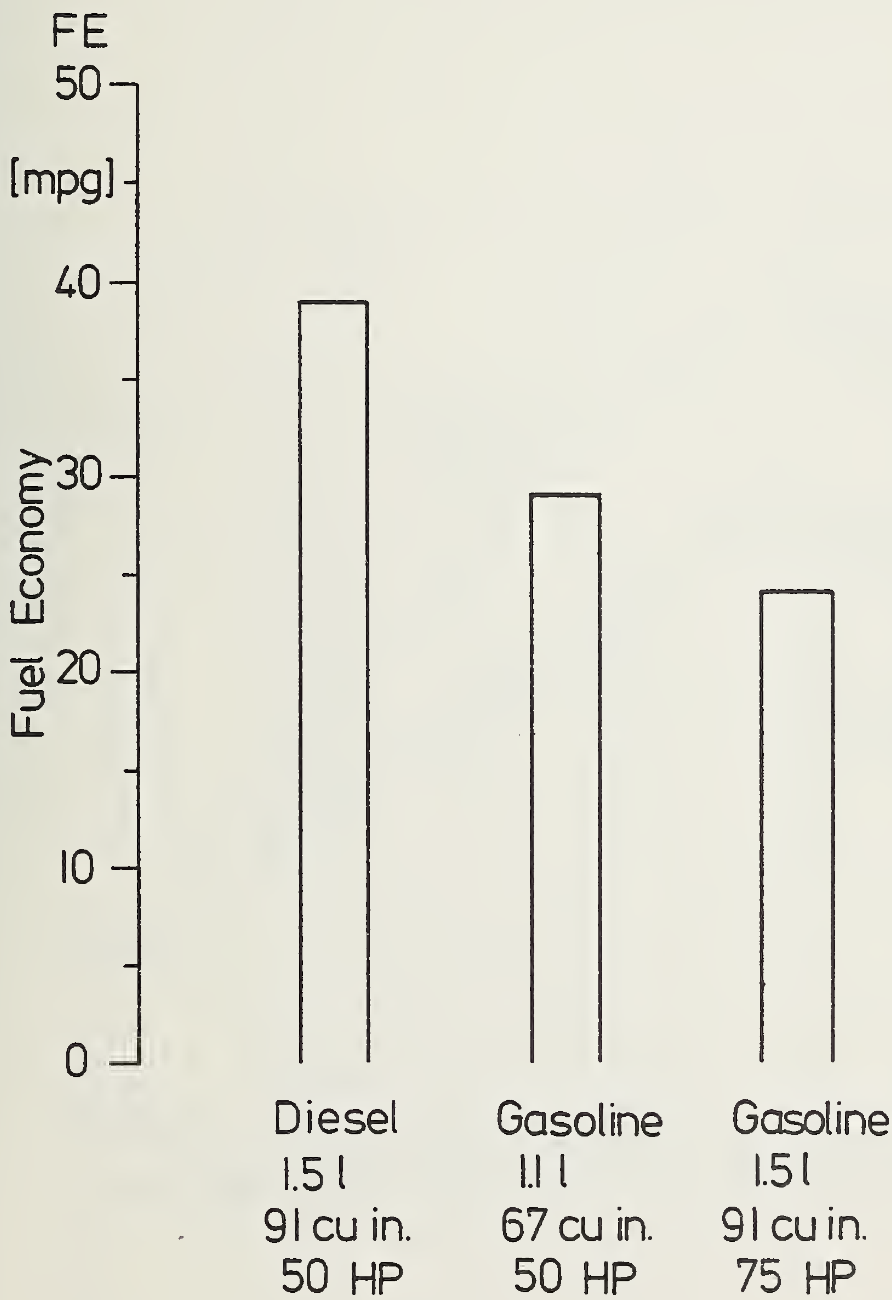

Comparison of Emissions
CO
[gpm]
23.17
9.0
7.0
3.4
1.32
HC
[gpm]
2.22
0.2
0.41
0.41
0.41
US TEST 75
NOx
[gpm]
2.76
2.0
1.2
1.0
1.5
1.0
0.4
* 1977 - 1979
1980 + 1981
1982
Gasoline
1.1 l / 50 HP
Diesel
1.5 l / 50 HP
Limit
1980
Limit
from 1981
US-Federal
Limit
from *
California

Fig.: 3.3.5



FE
[mpg]
Fuel Economy
50
40
30
20
10
0
Diesel
1.5 l
91 cu in.
50 HP
Gasoline
1.1 l
67 cu in.
50 HP
Gasoline
1.5 l
91 cu in.
75 HP

Fig.: 3. 3. 6

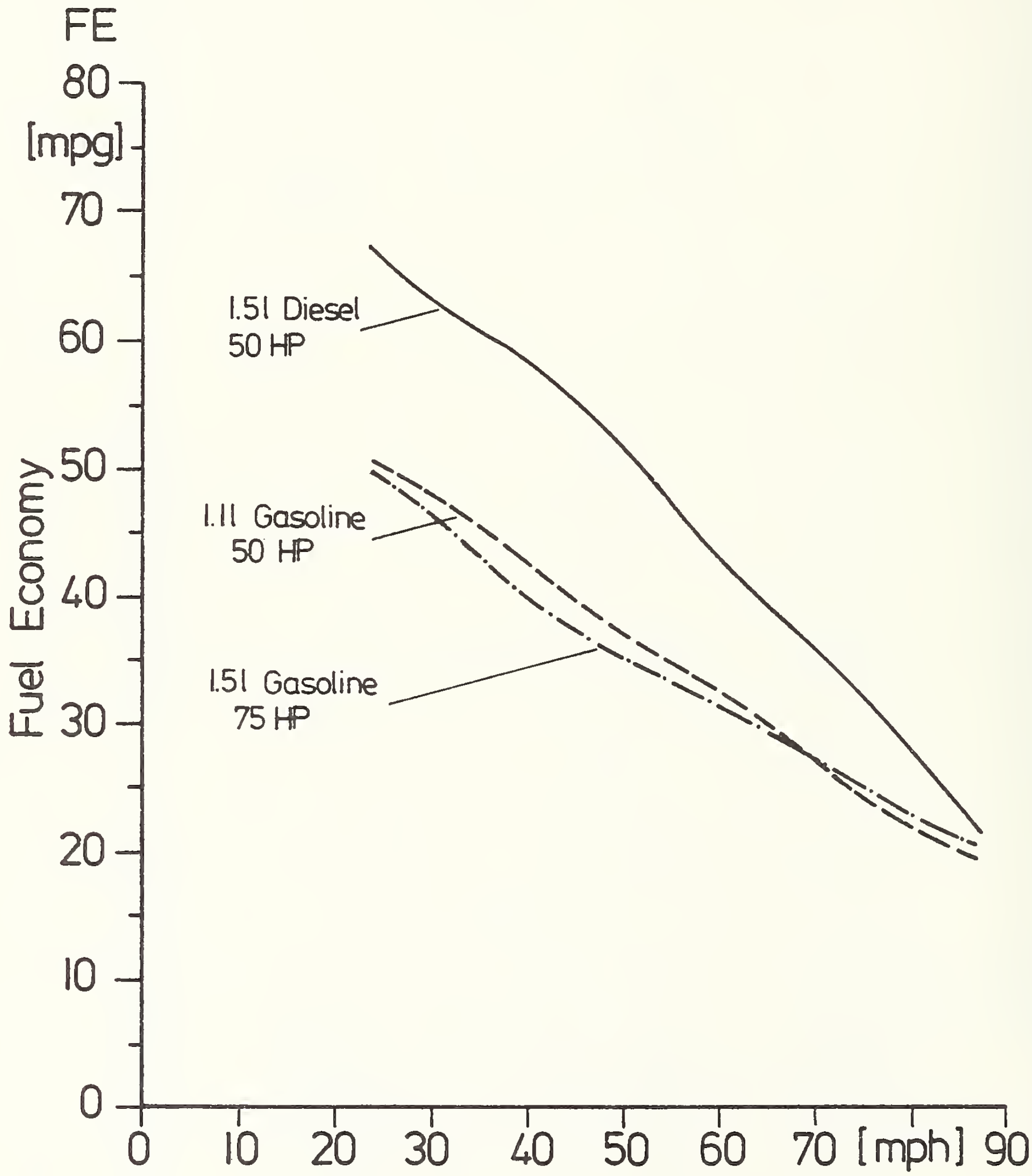

FE
80
[mpg]
70
60
50
40
30
20
10
0
Fuel Economy
1.5l Diesel
50 HP
1.1l Gasoline
50 HP
1.5l Gasoline
75 HP
0
10
20
30
40
50
60
70
[mph]
90

Fig.: 3.3.7

COMPARISON OF FUEL CONSUMPTION: DIESEL AND GASOLINE ENGINE

of Equal Performance

of Equal Displacement

Fuel Consumption Test Procedure	Gasoline 1.1 l 50 HP		Diesel 1.5 l 50 HP	Gasoline 1.5 l 75 HP	
Constant Speed (mph)	mpg	Increase in Fuel Consumption (%)	mpg	mpg	Increase in Fuel Consumption (%)
24.8	52.3	40.6	73.6	51.2	43.8
31.1	49.0	37.0	67.3	47.1	42.9
37.3	45.3	40.5	63.6	42.0	51.3
43.5	41.3	42.5	58.9	38.0	55.0
49.7	38.0	37.8	52.3	35.1	48.9
55.9	35.1	36.0	47.1	32.7	44.0
62.1	31.4	36.3	42.8	31.4	36.4
68.3	27.7	37.1	38.0	28.0	35.5
74.5	24.8	32.0	32.7	25.3	29.2
80.7	22.4	18.0	26.5	22.9	15.7
87.0	20.0	12.4	22.4	20.5	9.5
European Cycle	21.2	30.4	27.7	17.1	62.3
EPA	mpg		mpg	mpg	
Urban	29.0	34.5	39.0	24.0	62.5
Highway	42.0	23.8	52.0	37.0	40.5
Combined	34.0	29.4	44.0	28.0	57.1

Tab.: 3.3.1

4. THE ENGINES

In this Section, all engines considered in this Study will be described along with those which were available for measurements performed under the terms of this Contract. We decided not to discuss the question of engine system cost in this Chapter but to deal with it as a subdivision of Chapter 6.3 "Consumer Acceptability".

4.1 ENGINE OUTLAY AND DESIGN

4.1.1 Engine Design Parameters

Table 4.1.1 shows the engine design parameters of the engines analyzed in this Study.

The engines identified by Modification Codes have been tested under the terms of this Contract.

4.1.2 Emission Control Concepts

The engines which in Table 4.1.1 are numbered 1 - 3, 6 and 10 - 12 as well as Modification Codes 1 - 4 of engines 4, 7, and 9 were Uncontrolled according to the definition of the term given in the U.S. standards. They are engines with close manufacturing tolerances, intensive quality control, exact tuning, and regular tuning checks by the VW service organization.

The other engines incorporate a number of different emission control concepts, depending on which emission level or Engineering Goal had to be met.

Table 4.1.2 concentrates on the Controlled engines, listing their Emission Tasks or Engineering Goals as well as their control concepts which, especially in Modification Codes 13 - 20, were instrumental in meeting the Engineering Goals of the advanced standards.

4.1.3 Engine Parameter Variations

This Report refers to variations in the following parameters: Throttle angle, air/fuel ratio, spark advance, and amount of exhaust gas recirculated. Some of the data were taken from outside sources, some were

generated by VW in connection with other projects hitherto unpublished, and some were generated specifically for this Report.

One of the major features of our work under this Contract was to generate mean effective pressure vs. engine speed maps of the following engine modifications:

01/03/04/09/11/ to 13 and 15 to 20.

All engine map measurements were made without the use of catalysts because we were interested in raw emissions. Furthermore, some modifications are based on vehicle changes only, that is why the engines disregarding catalysts are the same in the Modification Codes given below.

The engine map of modification code

01 corresponds to those of 02 and 05;

03 corresponds to 07;

09 corresponds to those of 06 and 10;

12 corresponds to 08; and

13 corresponds to 14.

For this reason, the tests given above suffice to generate engine maps for all Modifications. These engine maps feature the following:

Lines of constant throttle opening angle and power output;

Lines of constant specific fuel consumption expressed in g/HPh and g/kWh;

Lines of constant absolute fuel consumption expressed in l/h and kg/h;

Lines of constant CO emission expressed in % and g/h;

Lines of constant HC emission expressed in ppm and g/h;

Lines of constant NO emission expressed in ppm and g/h;

Lines of constant O_2 emission expressed in % and g/h;

Lines of constant CO_2 emission expressed in % and g/h;

Lines of constant spark advance; and

Lines of constant air/fuel ratio.

All these engine maps are contained in the form of diagrams, and of data blocks in Appendix.

These engine maps have been used to compare individual engines as well as to project UDC emissions, HDC emissions, and fuel economy.

The engine map configuration of an engine constitutes a decisive influence on the fuel economy of the vehicle concerned. Engine maps should be designed so that all ranges frequently used under urban and highway traffic conditions are carefully geared to maximum fuel economy. However, in order to safeguard good startability, good idle conditions driveability, and acceleration performance it is often impossible to adjust the low-power, low-speed- and full-power, and high-speed areas to maximum fuel economy, although this would influence fuel consumption only a little, if WOT-operation is concerned. But in case of warm-up conditions and idle there is an important influence on fuel economy.

No	Modification Codes	No. of Cylinders	Displacement cm³	Displacement in³	Bore [mm]	Stroke [mm]	Compression Ratio	Combustion Chamber	CAMSHAFT Intake valve opens bTDC	CAMSHAFT Intake valve closes aBDC	CAMSHAFT Intake valve max. lift mm	CAMSHAFT Exhaust valve opens bBDC	CAMSHAFT Exhaust valve closes aTDC	CAMSHAFT Exhaust valve max. lift mm	Carburetor	Fuel Injection
1	not tested under this contract	4	895	54.6	69.5	59	8.2		0°	31°	9	35°30'	bTDC 7°30'	8.5	×	
2	"	"	1092	66.6	"	72	8.0		2°	38°	"	41°	bTDC 3°	"	×	
3	"	"	1272	77.6	75	"	8.2		7°	42°	10	51°	bTDC 4°	9.6	×	
4	03/07/11	"	"	"	"	"	"		"	"	"	"	"	"	×	
5	15/19	"	"	"	"	"	"		"	"	"	"	"	"		×
6	not tested under this contract	"	1457	88.9	79.5	74.5	8.0		4°	46°	10.3	44°	6°	10.3	×	
7	01/02/05/06/09 10	"	1588	96.9	"	80	"		"	"	"	"	"	"	×	
8	13/14/17/18	"	"	"	"	"	"		"	"	"	"	"	"		×
9	04/08/12	"	"	"	"	"	8.2		7°	54°	10.7	51°	10°	10.7	×	
10	not tested under this contract	"	"	"	"	"	9.3		4°	46°	10.3	44°	6°	10.3		×
11	"	"	1605	97.9	82	76	9.5		5°	35°	8.2	36°30'	4°	8.3	×	
12	"	"	1807	110.3	87	"	"		6°	42°	9	42°	6°	9.2	×	
13	16/20	5	2144	130.8	79.5	86.4	8.0		"	44°	9.3	40°	10°	9.3		×

Tab.: 4.1.1

Modification Code	Emission Tasks [g/mi] HC	Emission Tasks [g/mi] CO	Emission Tasks [g/mi] NOx	Engineering Goals [g/mi] HC	Engineering Goals [g/mi] CO	Engineering Goals [g/mi] NOx	Number of cylinders	Displacement [cu in]	Combustion Chamber: Compression ratio nominal	Compression ratio maximum	Compression ratio minimum	Combustion Chamber: surface/Volume Ratio [in⁻¹]	Peak Horsepower at RPM	Max. Torque [kp×m] at RPM	CID / Hp	Carburetor with automatic(a)/manual (m) choke	Fuel Injection K-Jetronic	Deceleration control	Spark Control: Spark advance diaphragm	Spark Control: Spark retard diaphragm	Secondary Air System: Pump	Secondary Air System: self aspirating	EGR-Control: On-Off	EGR-Control: Proportional	Catalytic System: Ox cat 3"	Catalytic System: Ox cat 6"	Catalytic System: 3-way 6"	closed loop
01	uncontrolled			–	–	–	4	96.9	8.0	8.2	7.6	5.48	66.7 5400	11.5 3600	1.453	a			×									
02	"			–	–	–	"	"	"	"	"	"	"	"	"	a			×									
03	"			–	–	–	"	77.6	8.2	8.4	8.0	6.37	60.4 5700	9.4 4200	1.285	m			×									
04	"			–	–	–	"	96.9	"	"	"	5.48	84.9 5800	12.6 4200	1.141	a			×									
05	1.5	15	3.1	1.0	10.0	2.1	4	96.9	8.0	8.2	7.6	5.48	66.7 5400	11.5 3600	1.453	a			×	×		×			×			
06	"	"	"	"	"	"	"	"	"	"	"	"	67.6 5600	11.9 2800	1.433	a		×	×	×		×	×		×			
07	"	"	"	"	"	"	"	77.6	8.2	8.4	8.0	6.37	60.4 5700	9.4 4200	1.285	m			×	×		×			×			
08	"	"	"	"	"	"	"	96.9	"	"	"	5.48	84.3 5800	12.3 4400	1.149	a		×	×	×		×	×		×			
09	0.9	9	2.0	0.6	6.0	1.4	4	96.9	8.0	8.2	7.6	5.48	67.6 5600	11.9 2800	1.433	a		×	×	×		×	×			×		
10	"	"	"	"	"	"	"	"	"	"	"	"	"	"	"	a		×	×	×		×	×			×		
11	"	"	"	"	"	"	"	77.6	8.2	8.4	8.0	6.37	55.8 5600	9.0 3200	1.391	m		×	×	×		×	×			×		
12	"	"	"	"	"	"	"	96.9	"	"	"	5.48	84.3 5800	12.3 4400	1.149	a		×	×	×		×	×			×		
13	0.41	3.4	1.0	0.2	1.7	0.25	4	96.9	8.0	8.2	7.6	5.48	73.8 5800	11.2 3200	1.313		×	×	×	×	×					×	×	×
14	"	"	"	"	"	"	"	"	"	"	"	"	"	"	"		×	×	×	×	×			×		×	×	×
15	"	"	"	"	"	"	"	77.6	8.2	8.4	8.0	6.37	58.9 6000	9.1 3400	1.317		×		×	×							×	×
16	"	"	"	"	"	"	5	130.8	8.0	8.3	7.7	4.88	101.7 5400	16.0 4200	1.286		×	×	×	×	×			×		×	×	×
17	0.41	3.4	0.4	0.2	1.7	0.1	4	96.9	8.0	8.2	7.6	5.48	72.5 5700	11.3 3300	1.337		×	×	×	×	×					×	×	×
18	"	"	"	"	"	"	"	"	"	"	"	"	74.1 5800	10.9 3600	1.308		×	×	×	×	×			×		×	×	×
19	"	"	"	"	"	"	"	77.6	8.2	8.4	8.0	6.37	58.8 5800	8.9 3400	1.320		×	×	×	×							×	×
20	"	"	"	"	"	"	5	130.8	8.0	8.3	7.7	4.88	102.6 5400	16.0 4200	1.275		×	×	×	×	×			×		×	×	×

Tab.: 4.1.2

4.2 ENGINE FAMILIES

This Chapter will deal only with those engines which were tested under this Contract.

4.2.1 Description of Engine Families

Table 4.1.2 (see Chapter 4.1) gives a summary of the emission control concepts applied to the various Modification Codes.

In order to highlight their function somewhat better, the engines and their control concepts are shown schematically in Figs. 4.2.1 through 4.2.7. There is a list attached in Tab. 4.2.1 enumerating the various devices used.

To facilitate understanding, Fig. 4.2.8 shows the principle of a fuel distributor without and with Lambda control, a device which is used in Modification Codes 13 - 20.

The fuel metering slits (V) - there is one for each cylinder - are very carefully made. They are controlled by an air flow sensor, which opens or closes them depending on the air flow. In other words, in a space where there is a pressure of p_1 the quantity of fuel emitted is the same in all four cases.

Pressure (p_1) acts on a diaphragm (M) along with a spring (S). From the other direction, pressure p_2, which corresponds to the initial pressure of the fuel pump (Dp), acts on the diaphragm as well. A balance between the force of the spring S and the pressure p_1 on one side and pressure p_2 on the other side is reached as soon as the diaphragm has yielded to a certain degree, uncovering a ring-shaped aperture at the lower end of pipe H through which the fuel can escape and proceed to the inlet valves of the engine via injection nozzles.

When operating in connection with a Lambda control system, this unit functions in a rather similar way.

The signal of the Lambda sensor affects p_2 alone. Here, pressure p_2 is not merely governed by the fuel line pressure behind the pump but also by the frequency with which a pulse valve opens and closes. The longer the valve remains open, the lower is p_2, and the more fuel will flow towards the cylinders. If the pulse valve remains closed most of the time, p_2 increases, and the flow of fuel to the cylinders is restricted.

Fig. 4.2.9 shows the system of EGR control and amplification which controls the quantity of exhaust gas recirculated in proportion to the air flow.

The air throughput is established by tapping the vacuum from the narrowest region of a venturi tube and conveying it to a large diaphragm (1). The diaphragm lifts, and a valve (2) opens, exposing a tube (3) in which there is a vacuum transmitting it to cavity (4). This vacuum, which is generated in the intake manifold, acts on the diaphragm of the EGR valve (5), opening it, and on the little diaphragm (6), until a balance is established between diaphragms 1 and 6. As the venturi vacuum grows, a new balance is struck, because the vacuum in cavity (4) grows as well, which means that an EGR quantity corresponding to the new venturi vacuum is conveyed to the intake manifold. Only if the throttle opens at a certain wide angle (WOT operation) does the venturi vacuum grow so intensive that it becomes more powerful than the intake manifold vacuum. At this point, valve (7) opens, permitting the low intake manifold vacuum to bypass the ball check valve (8), which was closed when manifold vacuum decrease and to act on the EGR control valve, which then closes as well.

Once the intake manifold vacuum drops below the venturi vacuum the EGR valve opens again. If the venturi vacuum continues to drop because of the throttle being closed the balance struck between diaphragms 1 and 6 is such that the valve plate (2) is in contact with its inner but not with its outer seat, so that ambient pressure can reach cavity (4), the effect of which is that the EGR valve is closed even further.

Except for the WOT operation range, where EGR is not wanted in spite of a high venturi vacuum, the quantity of recirculated exhaust gas passing the EGR valve is always proportional to the venturi vacuum.

4.2.2 Fuel Requirements

Under the terms of this Contract no effort was made to find any borderline fuels, i.e. those fuels which are just about good enough to be accepted by the engines under consideration. We merely established whether or not all engines will run on commercial U.S. fuels, which we found to be the case. The uncontrolled engines will run on leaded regular fuel, whereas the controlled engines with catalysts use unleaded fuel.

Tables 4.2.2 and 4.2.3 list the Federal specifications covering leaded and unleaded fuels as well as the specifications of the leaded and unleaded fuels used by us.

In all fuel economy and exhaust emission tests the fuels listed in Table 4.2.2 were used, whereas in the mileage accumulation of 1977 the fuels of Table 4.2.3 were used.

4.2.3 Requirements for Non-Standard Materials

The only cases where non-standard materials occur in our emission control concepts are those of the 3-way catalyst and the Lambda sensor.

Three-Way Catalyst

In 3-way catalysts, rhodium and ruthenium may be used as alternatives to platinum. Ruthenium does not raise any availability problems, but it is feared that the element may be unstable. This is of outstanding significance because ruthenium oxides are toxic, the most dangerous oxide being RuO_4. For this reason, automobile manufacturers using ruthenium are required to establish the stability of the element to the satisfaction of the emission authorities. For emission reasons, the use of ruthenium in connection with twin-bed concepts is especially advantageous because tests have shown that ruthenium catalysts form less NH_3 than rhodium catalysts.

There are no availability problems as regards rhodium either, provided that catalysts are made to a platinum/rhodium ratio of 19 : 1, which is the quantitative ratio of the natural resources of these two metals. However, all findings available at present indicate that it will be impossible to meet the stringent emission standards of 1981 with a platinum/rhodium ratio of 19 : 1. Still, there are two considerations justifying the use of 3-way catalysts manufactured to platinum/rhodium ratios of less than 19 : 1.

The first consideration is that precious metals, and especially rhodium, may be recycled. Rhodium may be salvaged from used catalysts at estimated rate of 70 to 80 %.

The second consideration is the possible use of 3-way catalysts in a two-bed system, the basic idea being to balance the platinum/rhodium ratio of the 3-way catalyst and the platinum/palladium ratio of the oxydation catalyst so that the overall platinum/rhodium ratio of the two-bed system as a whole amounts to 19 : 1. This condition would be met, for instance, if the 3-way catalyst were to have a platinum/rhodium ratio of 11 : 1, and the oxydation catalyst a platinum/palladium ratio of 5 : 2. The end result would be a platinum/rhodium ratio of 19 : 1.

In the tests run under this Contract, the platinum/rhodium ratio of the 3-way catalysts was 5 : 1, and that of the oxydation catalysts was 12 : 1.

Lambda Sensors

The only material used in Lambda sensor whose availability is limited is yttrium oxide. However, the quantities of yttrium oxide required are so infinitesimal that we do not expect any difficulties in that sector.

4.2.4 Weights and Volumes of Engine Systems

Table 4.2.4 enumerates the major dimensions of the engine systems under consideration. The engine dimensions listed in this Table signify the minimum interior dimensions of a rectangular crate which will hold the engine in question.

By reductions in scale especially in width, but also in length and height we indicated to what extent certain engine parts project beyond the dimensions given. Projections of this kind give a somewhat distorted picture of the engine compartment space required by an engine, because they frequently can be accomodated without any trouble at all as they occur only in strictly limited sectors of the engine and do not really require any significant enlargement of the engine compartment.

In designing an engine, catalysts can generally be located more or less at random, it being necessary only to instal them at the proper distance from the outlet valve. The same consideration applies to the fuel injection system and air filter in fuel injection engines. For this reason, our engine dimensions do not incorporate any equipment of this kind, the dimensions of which are listed separately. This applies particularly to the fuel injection system whose dimensions are listed in Table 4.2.5.

Table 4.2.6 shows the masses of the Modification Codes. Here again, we have introduced some kind of substructure to render the totality of data a bit more transparent. In this Table, the figures entered in boxes represent the total weight of a system. All figures have been rounded off to 1 kg. The maximum additional weight of equipment such as tubes, little valves, etc. may amount to another kilo.

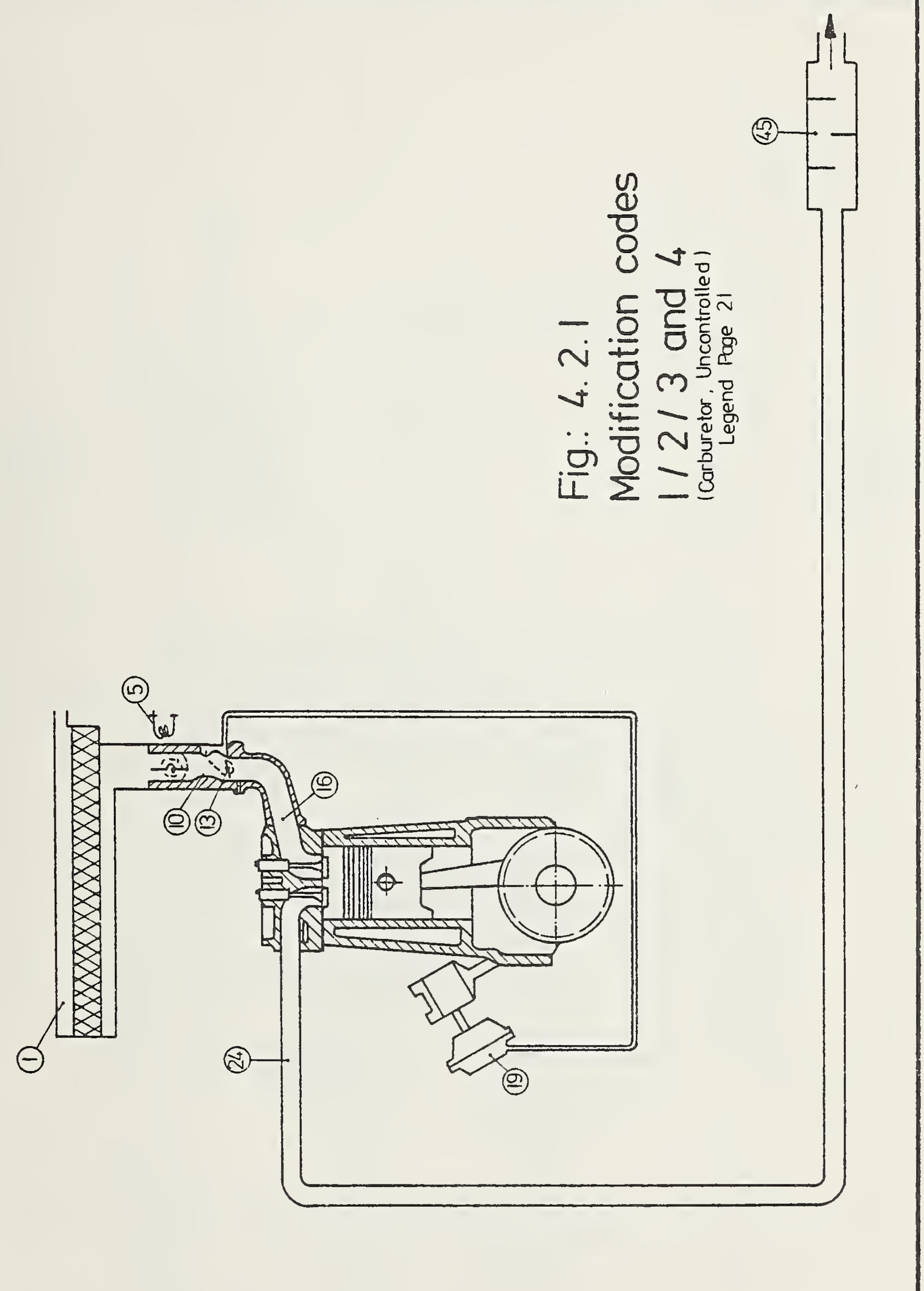


Fig.: 4.2.1
Modification codes
1/2/3 and 4
(Carburetor, Uncontrolled)
Legend Page 21

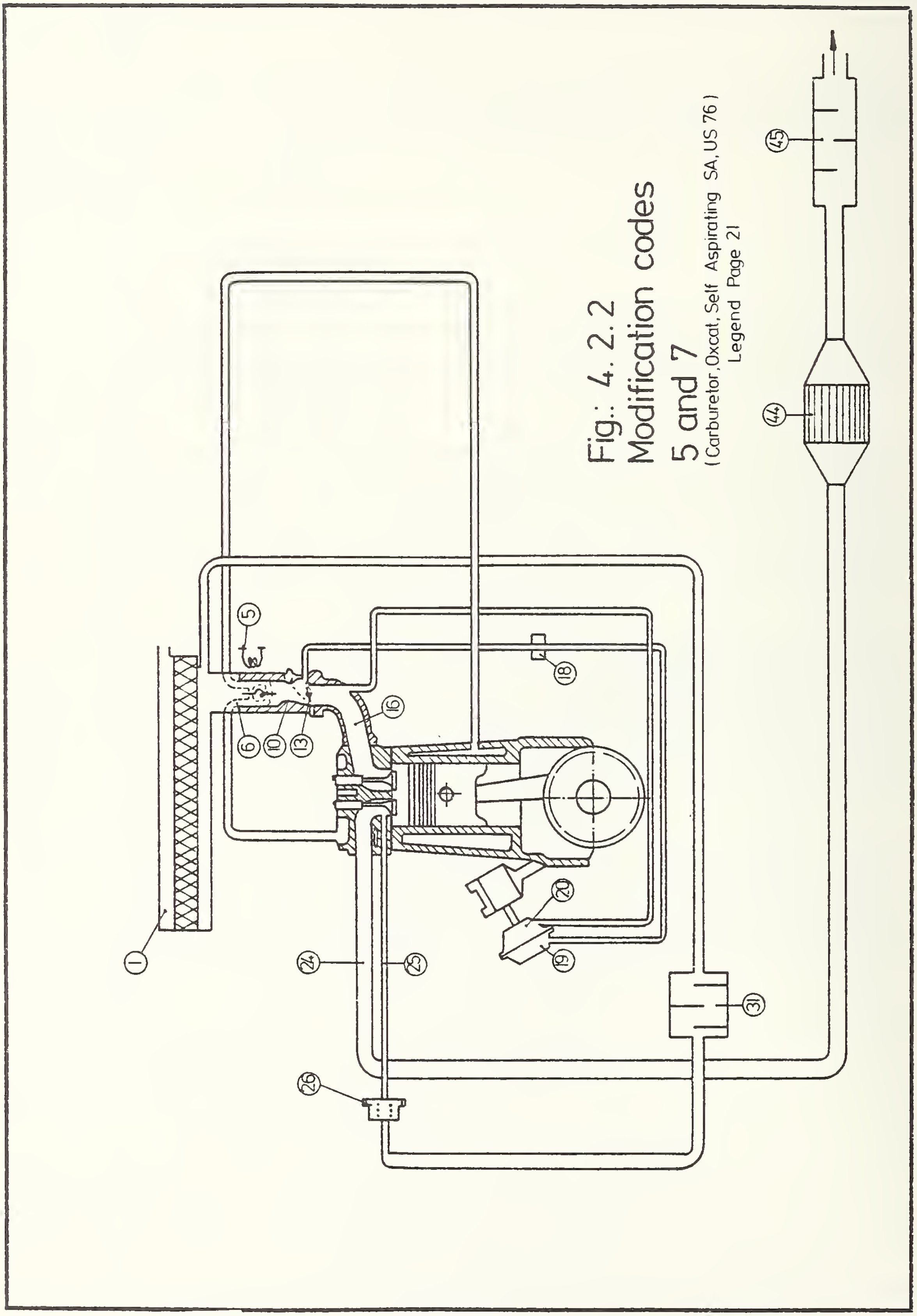


Fig.: 4. 2. 2
Modification codes
5 and 7
(Carburetor, Oxcat, Self Aspirating SA, US 76)
Legend Page 21

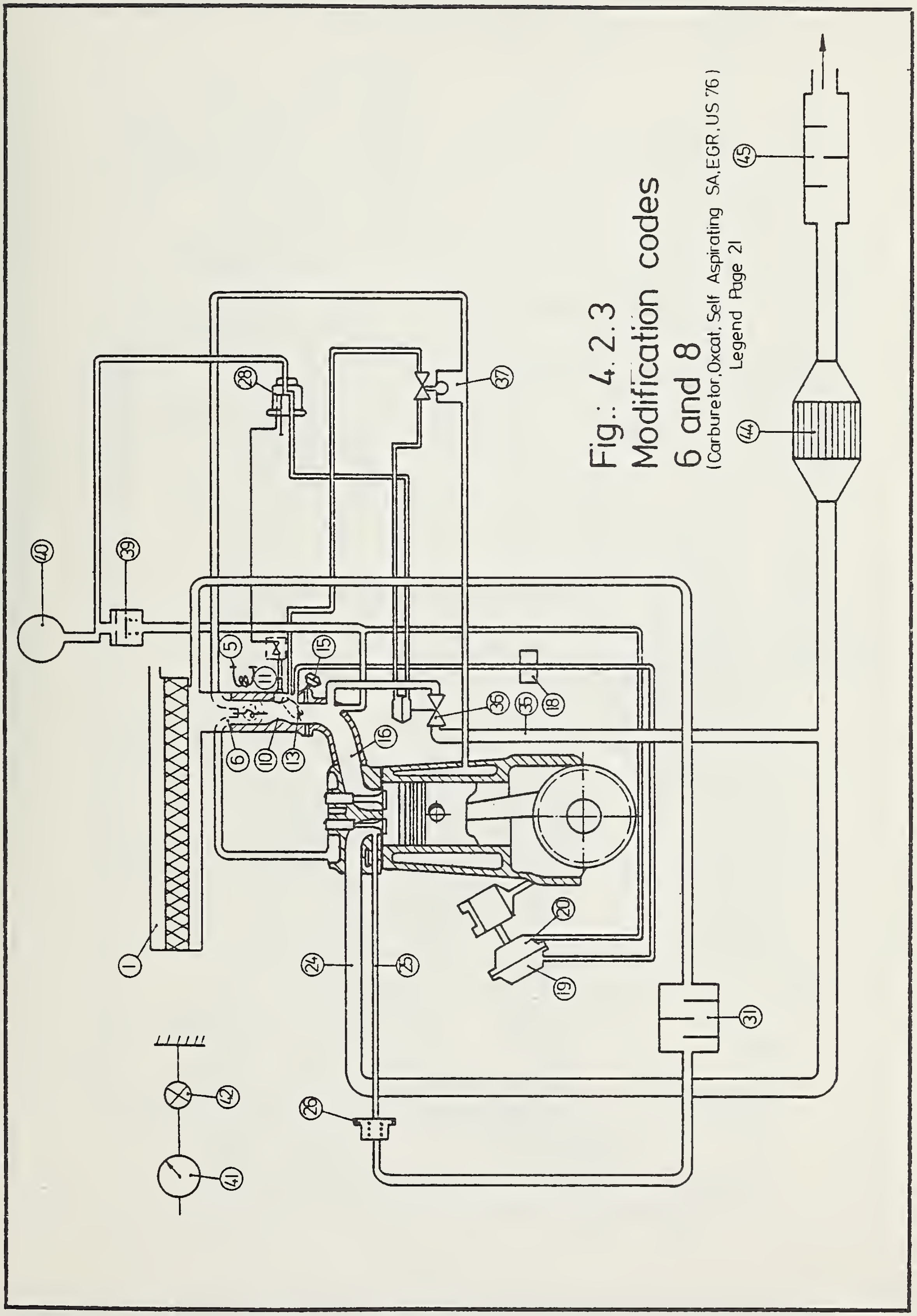


Fig.: 4.2.3
Modification codes
6 and 8
(Carburetor, Oxcat, Self Aspirating SA, EGR, US '76)
Legend Page 21

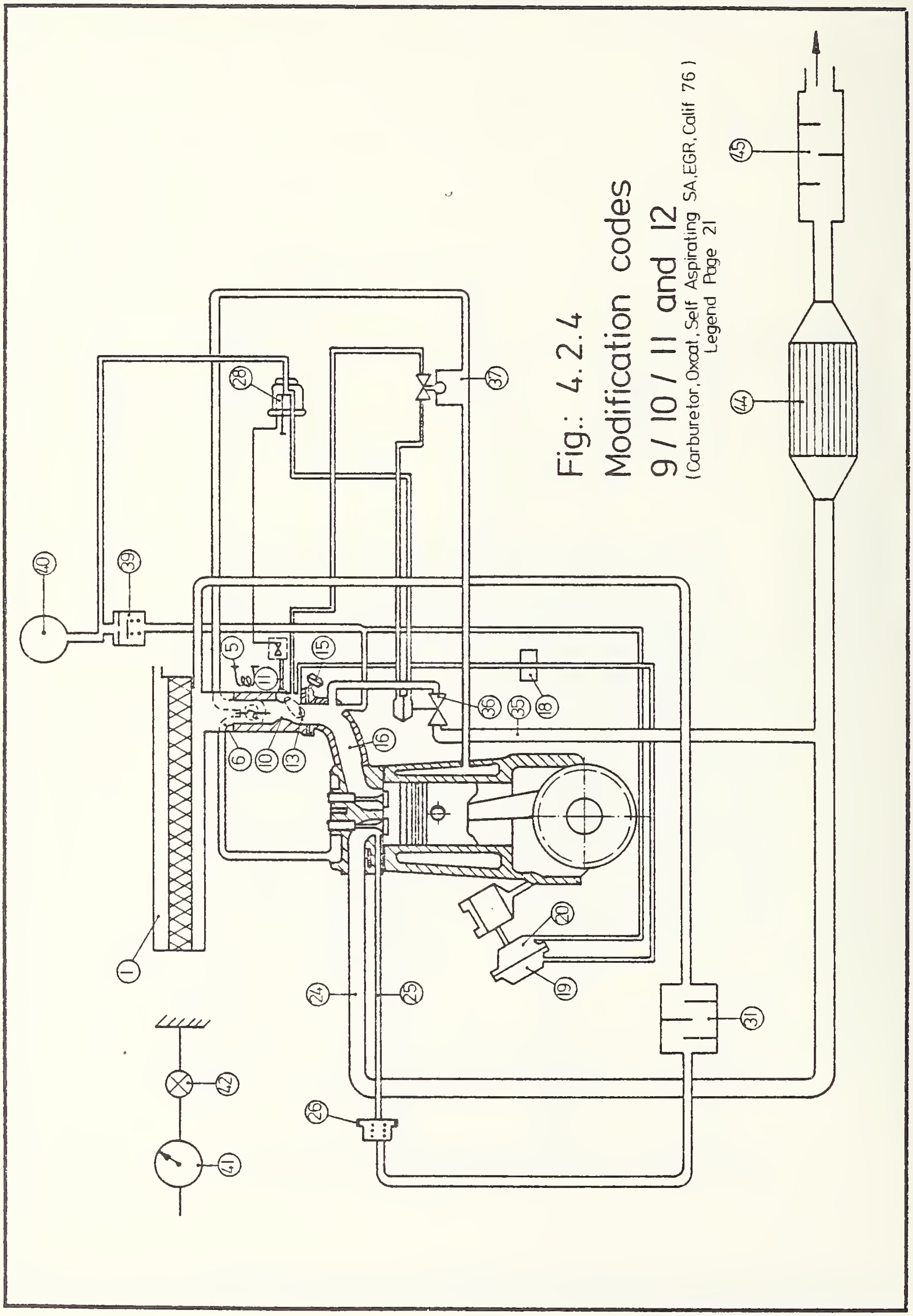


Fig.: 4.2.4
Modification codes
9 / 10 / 11 and 12
(Carburetor, Oxcat, Self Aspirating SA, EGR, Calif 76)
Legend Page 21

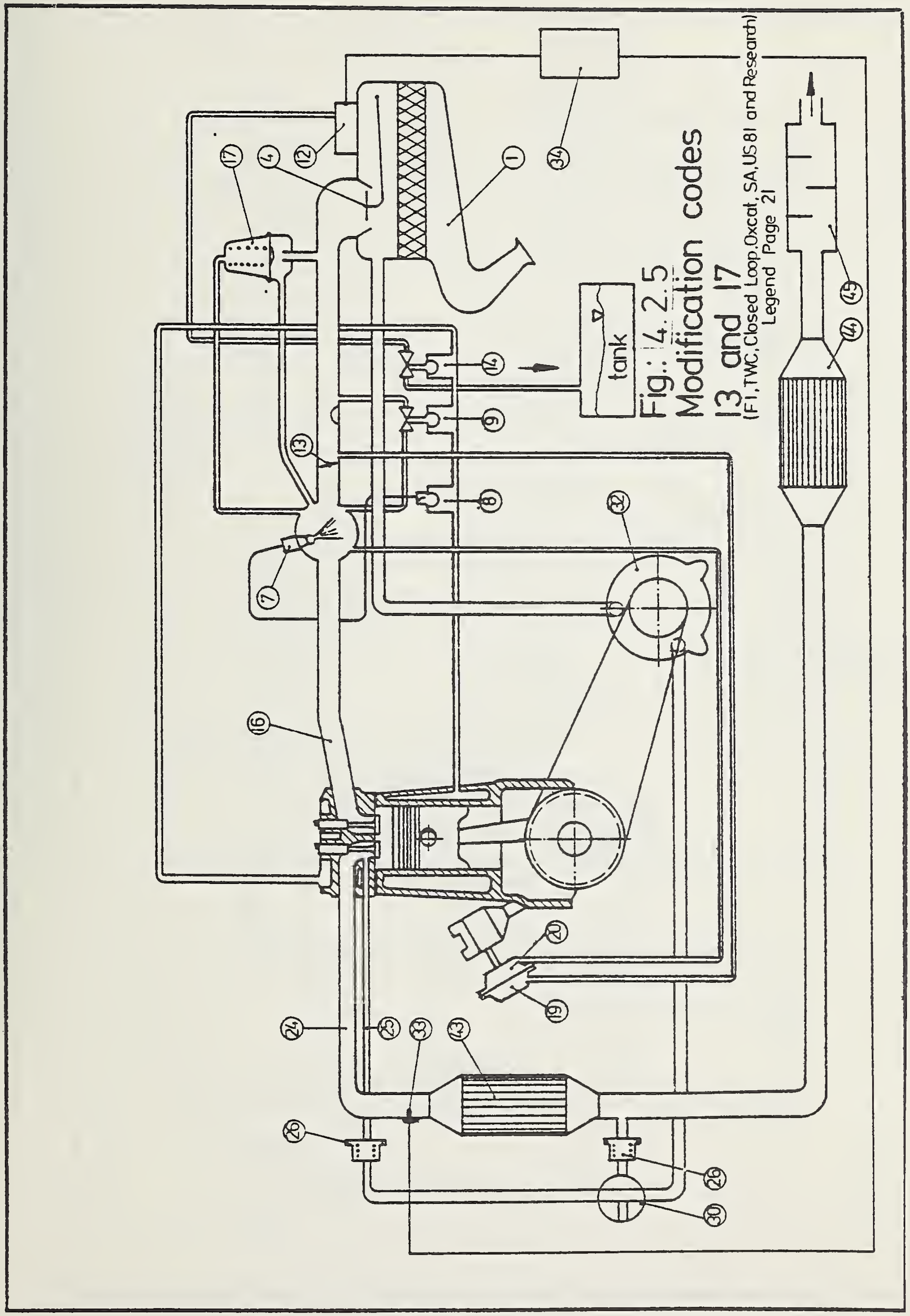


Fig.: 4.2.5
Modification codes
13 and 17
(F1, TWC, Closed Loop, Oxcat, SA, US 81 and Research)
Legend Page 21

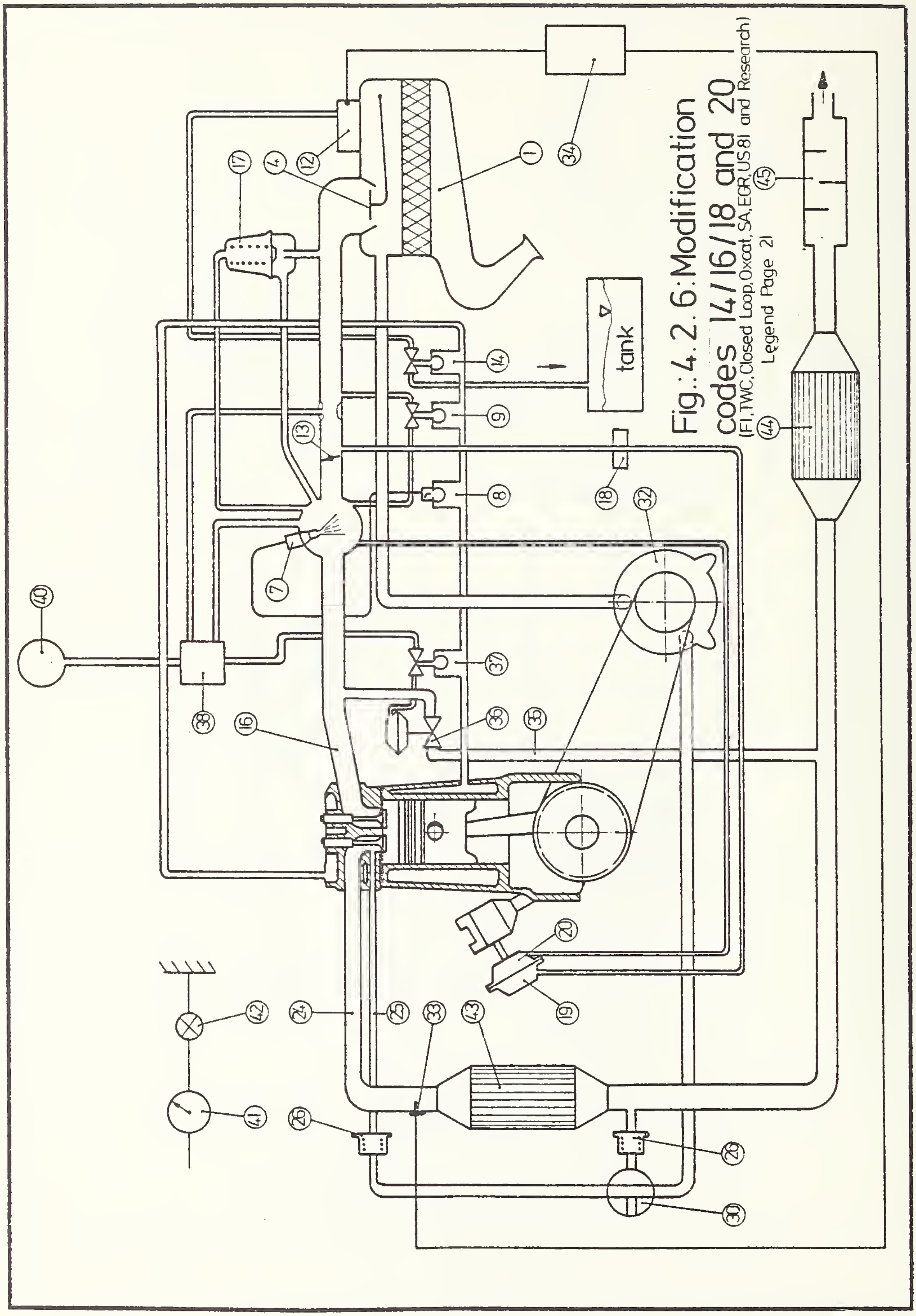


Fig.: 4.2.6: Modification codes 14/16/18 and 20
(FI, TWC, Closed Loop, Oxcat, SA, EGR, US 81 and Research)
Legend Page 2)

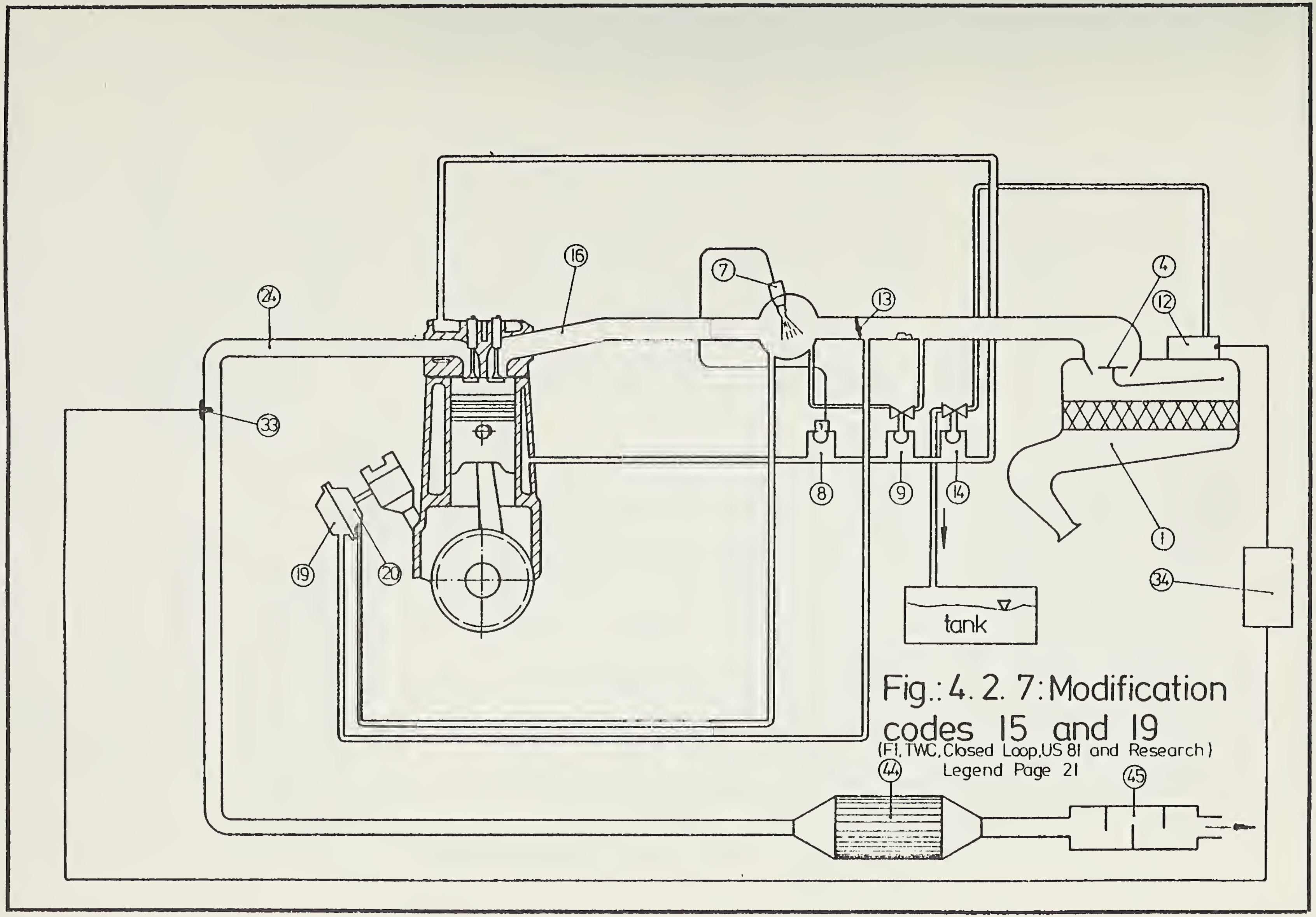
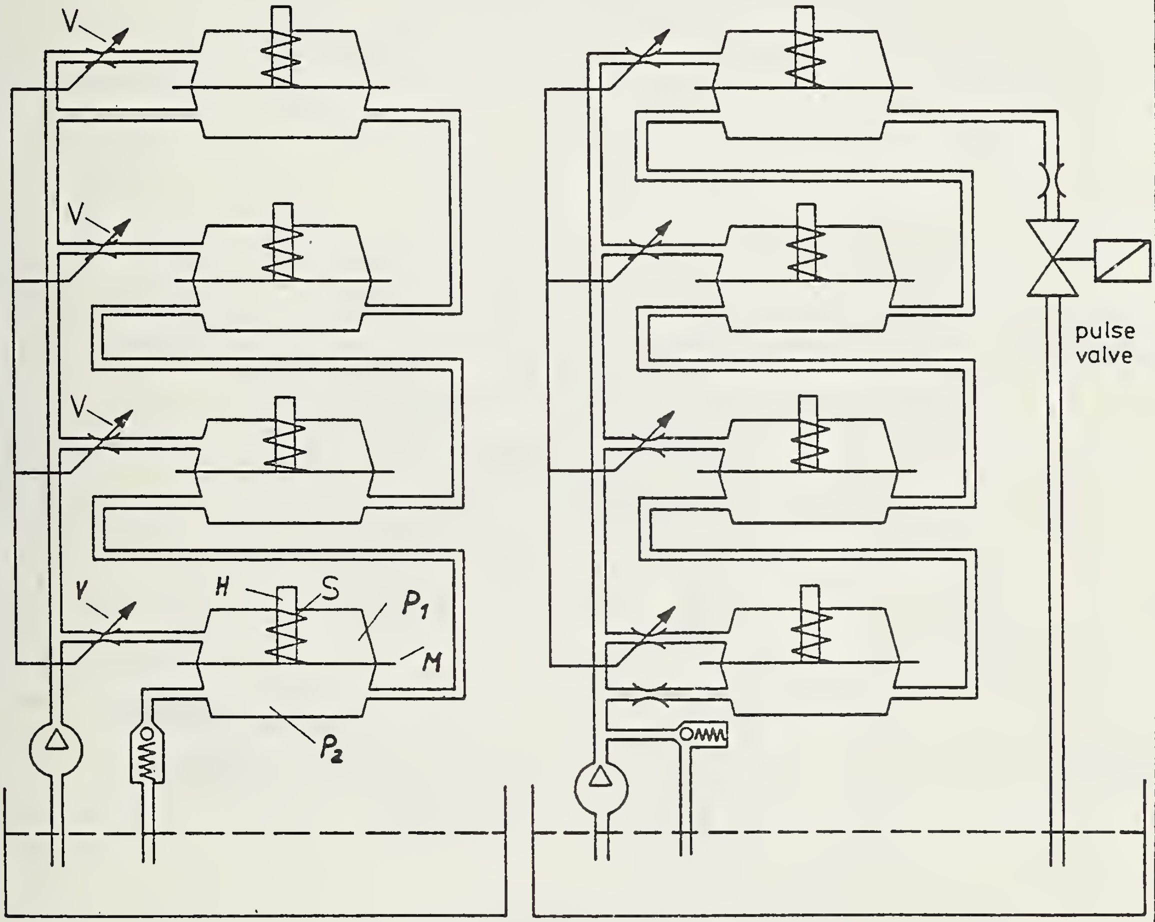


Fig.: 4. 2. 7: Modification codes 15 and 19
(FI, TWC, Closed Loop, US 81 and Research)
Legend Page 21

1	Air cleaner
4	Air flow sensor
5	Electrically heated choke system
6	Water heated choke system
7	Starting solenoid
8	Temperature related time switch
9	Auxiliary air valve
10	Carburetor
11	Wide open throttle switch
12	Fuel distributor
13	Throttle
14	Warm up regulator
15	Closing damper
16	Intake manifold
17	Deceleration control valve
18	Time delay valve
19	Spark advance diaphragm box
20	Spark retard diaphragm box
24	Exhaust pipe
25	Secondary air pipe
26	Secondary air check valve
28	Dual solenoid
30	Dual valve
31	Secondary air muffler
32	Secondary air pump
33	λ - sensor
34	λ - control unit
35	EGR cooling line
36	EGR valve
37	Thermal vacuum valve
38	EGR control amplifier
39	Vacuum check valve
40	Vacuum reservoir
41	Mileage counter for EGR
42	EGR check indication lamp
43	3-way catalyst
44	Oxidation catalyst
45	Muffler

Tab.: 4. 2. 1
Legend to
Fig.: 4.2.1 through 4.2.7



Fuel Distributor
without λ - control

Fuel Distributor
with λ- control

Fig.: 4. 2. 8

EGR-System Ranco

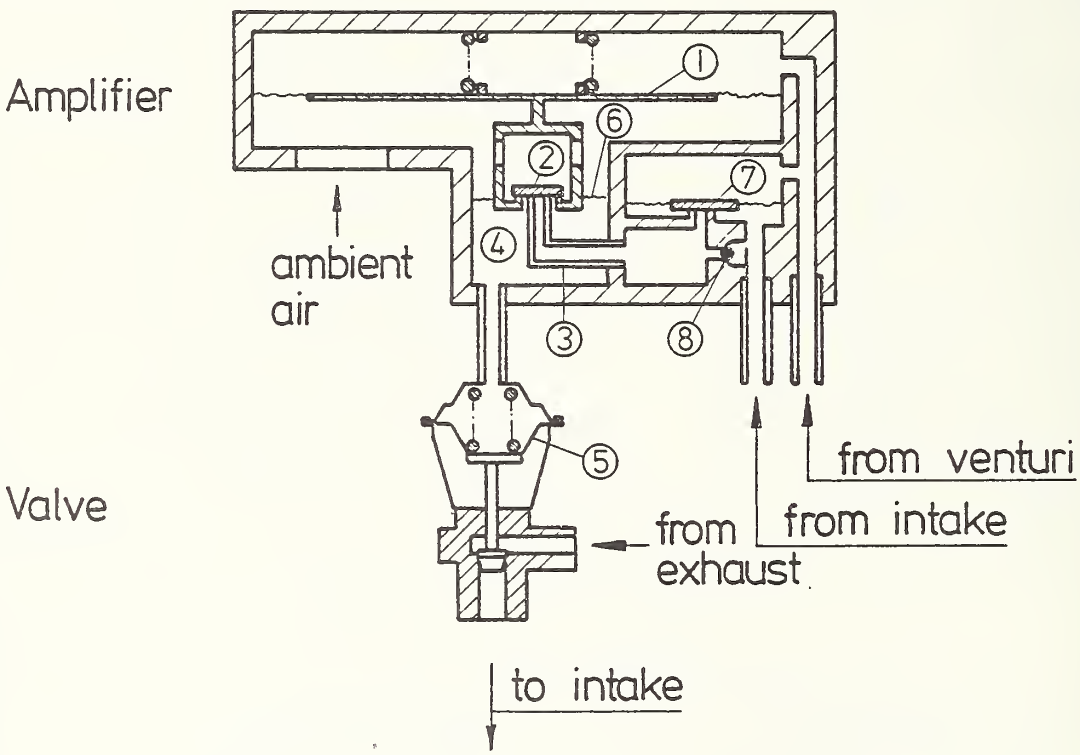


Fig.: 4.2.9

SPECIFICATIONS OF EXHAUST AND EVAPORATIVE EMISSION TEST FUEL FOR SPARK IGNITOIN ENGINES

Item	Federal Specifications		Specifications of Fuel Used	
	for Leaded Test Fuel	for Unleaded Test Fuel	Leaded	Unleaded
Octane, Research minimum	98	93	102	92
Sensitivity, minimum	9.0	7.5	10	10
Pb(Organic),gm/U.S.gal	1.4 min.	0.00 - 0.05	3.2	0.02
Distillation Range				
IBP °F°F	75 - 95	75 - 95	95	93
10% point °F	120 - 135	120 - 135	135	126
50% point °F	200 - 230	200 - 230	221	208
90% point °F	300 - 325	300 - 325	325	315
EP, max. °F	415	415	381	392
Sulfur, wt. %, max.	0.10	0.10	0.02	0.03
Phosphorus,gm./U.S.gal (max.)	0.01	0.005	0.004	0.0004
RVP, lb.	8.7 - 9.2	8.7 - 9.2	9.1	9.0
Hydrocarbon Composition				
Olefins, % max.	10	10	1	8
Aromatics, % max.	35	35	34	35
Saturates, %	Remainder	Remainder	Remainder	Remainder

Tab.: 4.2.2

SPECIFICATIONS OF MILEAGE ACCUMULATION TEST FUEL SPARK IGNITION ENGINES

Item	Federal Specifications for		Specifications of Fuel Used	
	Leaded Fuel	Unleaded Fuel	Leaded Fuel	Unleaded Fuel
Octane, Research (Range)	91 - 95	91 - 95	92	92
Minimum Octane (Research) Recommended	91	91		
Minimum Octane (Motor)			83	82
Pb. (Organic) gm/U.S.gal	1.4 min.		2.1	0.02
Phosphorus gm/U.S.gal			0	0.0004
Sulfur wt. percent			0.02	0.03
Hydrocarbon Composition				
Olefins, max. %			20	8
Aromatics, max. %			35	35
Saturates, %			Remainder	Remainder
RVP, lb. (Range) Summer/Winter	6.5 - 13	8.7 - 9.2	8.7 - 12.3	9.1
Additives				
Antifreeze			0.035 Vol.% Active Material	
Detergent			--	Shell M 400 (ASD) Active Material 100 ppm approx.

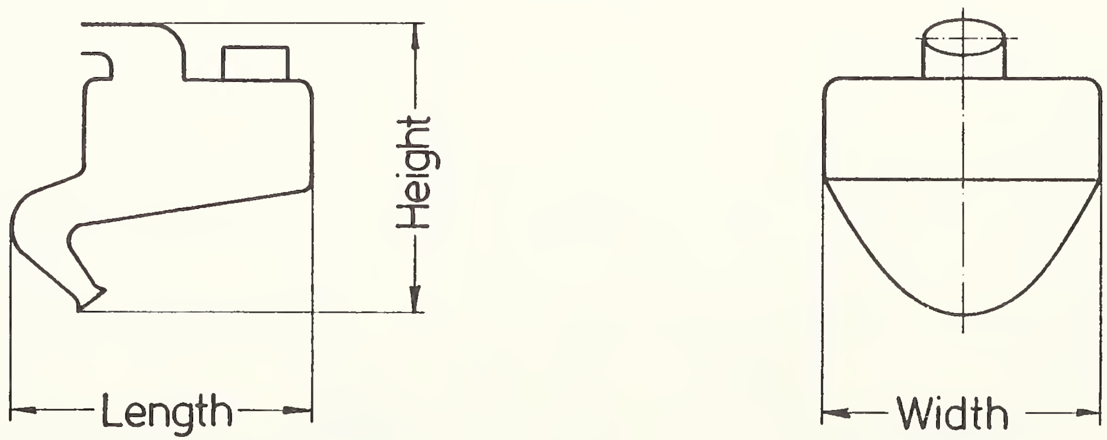
Tab.: 4.2.3

Modification Code	Width [cm] (Transversal) over: All	minus Distributor	minus Carburetor and Air Filter	minus Intake and Exhaust Manifold	minus Oil Filter	Length [cm] (Crankshaft Direction) over: All	minus Distributor	Height [cm] (Piston Direction) over: All	minus Carburetor and Air Filter	minus Intake Manifold	Additional Space Required for
1/2/4	60	→	→	41	37	54	→	70	61	→	——
3	67	→	48	39	→	61	53	63	→	→	——
5/6/8	60	→	→	41	37	54	→	70	61	→	Φ4"× 3"Cat
7	67	→	48	39	→	61	53	63	→	→	Φ4"× 3"Cat
9/10/12	60	→	→	41	37	54	→	70	61	→	Φ4"× 6"Cat
11	67	→	48	39	→	61	53	63	→	→	Φ4"× 6"Cat
13/14/17/18	54	→	→	41	37	54	→	70	→	61	2× Φ4"× 6"Cat Fuel Inject. System*
15/19	55	→	→	39	→	61	53	63	→	→	Φ4"× 6"Cat Fuel Inject. System*
16/20	50	48	→	→	→	72	→	83	→	79	Φ4"× 6"Cat Φ4,66"× 6"Cat Fuel Inject. System*

* = see Table 4.2.5

Tab.: 4.2.4

Weight and Volume of K-Jetronic System



Mod. Code	Width [cm]	Length [cm]	Height [cm]	Mass [kg]
13/14/15/17 18/19	19	34	33	7
16/20	32	43	34	8

Tab.: 4.2.5

Modification Code	Engine Mass [kg]	plus Intake Manifold [kg]	plus Secondary Air Pump and mounting [kg]	plus Carbureter and Air Filter [kg]	plus K-Jetronic System [kg]	plus Converter [kg]	plus Total of all Devices below 1 kg
1/2/4	109	110	——	$\boxed{113}$	——	——	——
3	95	96	——	$\boxed{99}$	——	——	——
5/6/8	109	110	——	113	——	$\boxed{115}$	x
7	95	96	——	99	——	$\boxed{101}$	x
9/10/12	109	110	——	113	——	$\boxed{116}$	x
11	95	96	——	99	——	$\boxed{102}$	x
13/14/17/18	109	113	116	——	123	$\boxed{129}$	x
15/19	95	98	101	——	108	$\boxed{111}$	x
16/20	163	168	172	——	180	$\boxed{187}$	x

Tab.: 4.2.6

5. VEHICLES ANALYSED

Depending on the requirements connected with the tests made under this Contract, the engines were either equipped with drivetrains, or the drivetrains were simulated, or the engines were installed in vehicles, or all vehicle data influencing fuel economy and emissions were simulated.

5.1 TRANSMISSIONS AND DRIVETRAINS

The data of all transmissions and drivetrains as well as those of the dynamic wheel diameters used in connection with the various Modification Codes are listed in Table 5.1.1.

In addition to vehicle dynamometer tests, a number of projections as well as UDC and HDC engine dynamometer tests involving a number of different drivetrains were performed. All data referring to these tests are given in Table 5.1.2. The transmissions are identified by letters.

Modification Code	m= manual a= automatic	Transmission Ratios Gears 1.	2.	3.	4.	Axle	Dyn. Wheel Diameter [cm]
01/02/05/06/09 10/13/14/17/18	m	3.46	1.94	1.29	0.97	3.90	55.8
03/07/11/15 19	m	3.46	2.05	1.35	0.96	4.06	54.8
04/08/12	m	3.46	1.94	1.29	0.91	4.44	60.0
16/20	m	3.60	2.13	1.36	0.97	4.11	60.0

Tab.: 5.1.1

Drivetrain No.	m = manual a = automatic	Transmission Ratios 1.Gear	2.Gear	3.Gear	4.Gear	5.Gear	Axle Ratio	Stall Speed [RPM]	Con-verter Torque Ratio	Dynamic Wheel Diameter [cm]
A	m	3.46	1.94	1.29	0.97	—	3.27	—	—	55.8
B	m	"	"	"	"	—	3.48	—	—	"
C	m	"	"	"	"	—	3.70	—	—	"
D	m	"	"	"	"	—	3.90	—	—	"
E	m	"	"	"	"	—	4.11	—	—	"
F	m	"	"	"	"	0.75	3.27	—	—	"
G	m	"	"	"	0.75	—	3.90	—	—	"
H	m	"	"	1.37	1.10	0.91	"	—	—	"
I	m	"	"	"	"	0.75	"	—	—	"
K	m	"	"	1.23	"	0.91	"	—	—	"
L	m	3.78	2.06	1.25	0.87	—	4.57	—	—	62.6
M	m	3.45	1.94	1.29	0.90	—	4.11	—	—	55.8
N	m	3.60	2.13	1.36	0.97	—	"	—	—	60.0
O	a	2.55	1.45	1.00	—	—	4.09	2000 - - 2450	2.44	62.6
P	a	"	"	"	—	—	3.76	2000 - - 2350	2.50	55.8
Q	a	"	"	"	—	—	3.91	"	"	"
R	a	"	"	"	—	—	"	1800 - - 2100	2.07	60.0

Tab.: 5.1.2

5.2 TEST VEHICLES AND ENGINES

The main vehicle data on which the fuel economy and emissions of the vehicles tested under the terms of this Contract depend are compiled in Table 5.2.1. Engines and transmissions can be allocated by means of the identification data given.

Modification Codes 02, 06, 10, 14,and 18 represent vehicles which do not exist in reality. There is no such thing as a Rabbit having an inertia weight of 3,000 lbs. Nevertheless, we tested the rabbit with all its data unchanged except for two different inertia weights because this is the easiest way to establish the extent of the influence of inertia weight. It is clear that the entire process is fictitious, mainly because a vehicle weighing 3,000 lbs and producing about 70 HP would naturally have transmission ratios different from those of a 2,250 lbs vehicle. The transmission ratios of a heavier vehicle would, of course, be higher, which would lead to an increase in fuel consumption. Yet there is nothing in the graphs of Chapter 6.1.3 to indicate that at an inertia weight of 3,000 lbs the fuel economy of Modification Codes 02, 06 and 10 tends to be better than that of Modification Codes 04, 08, and 12, to which alone it can be compared.

Table 5.2.2 is an overview of all main design parameters of all engines, drivetrains, and vehicles under their various Modification Codes with the resultant emission and fuel economy figures as well.

Fig. 5.2.1 shows the partial road load curves of the vehicles tested. The curves were generated by entering the torque required to keep a certain velocity in fourth gear over velocity or engine speed. The data used are listed in Table 5.2.3.

Fig. 5.2.2 shows the exact positions of the engine/vehicle systems of the various Modification Codes on the HP-HP/IW map. We can see that, contrary to the plan, it proved to be impossible to distinguish clearly between high and low HP/IW ratios, which is due to several peak horsepower deviations. This, however, is not so important, because we shall see a little further along (Chapter 6.1.4) that fuel economy is independent of the horsepower-to-inertia-weight ratio.

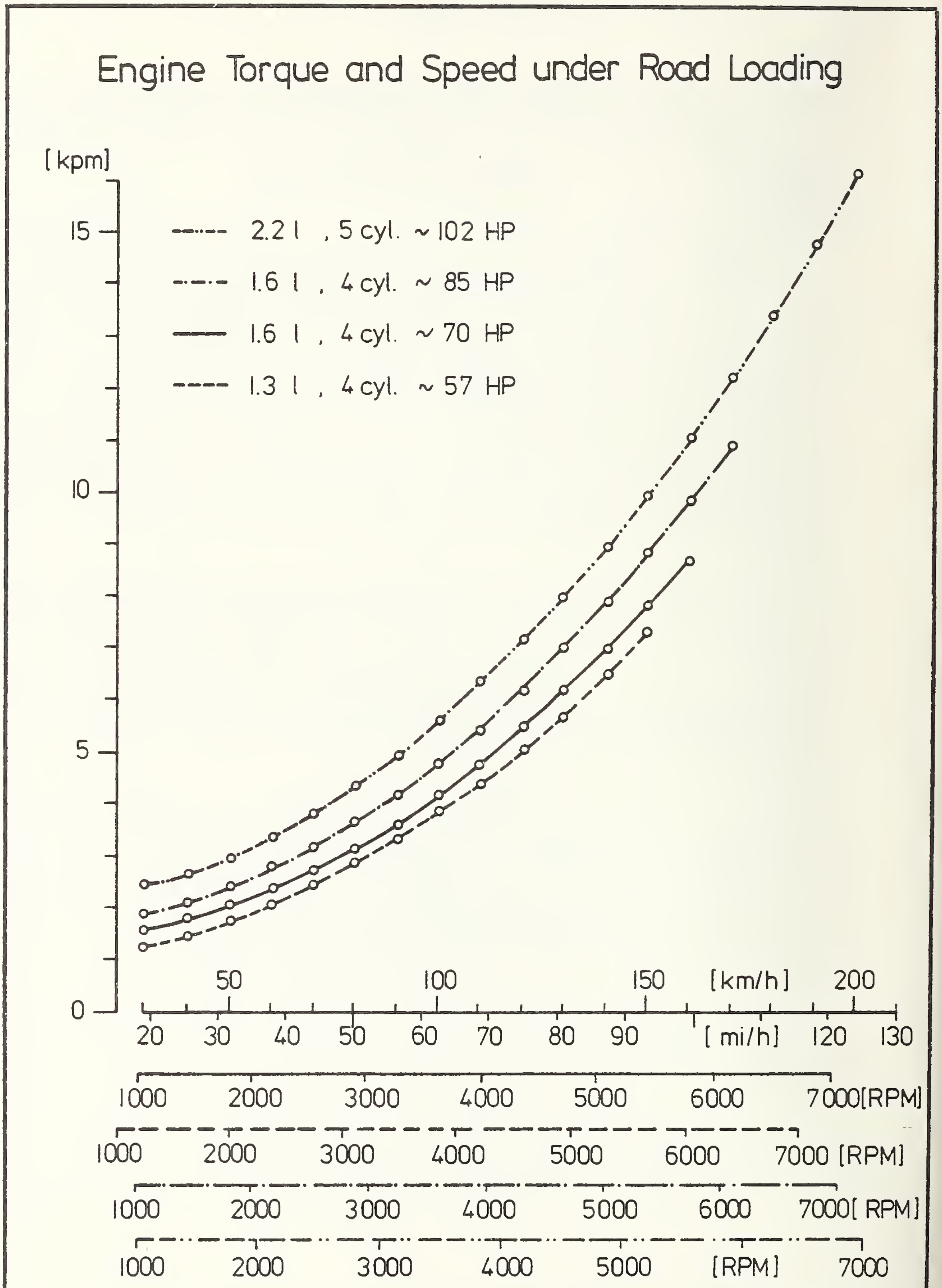
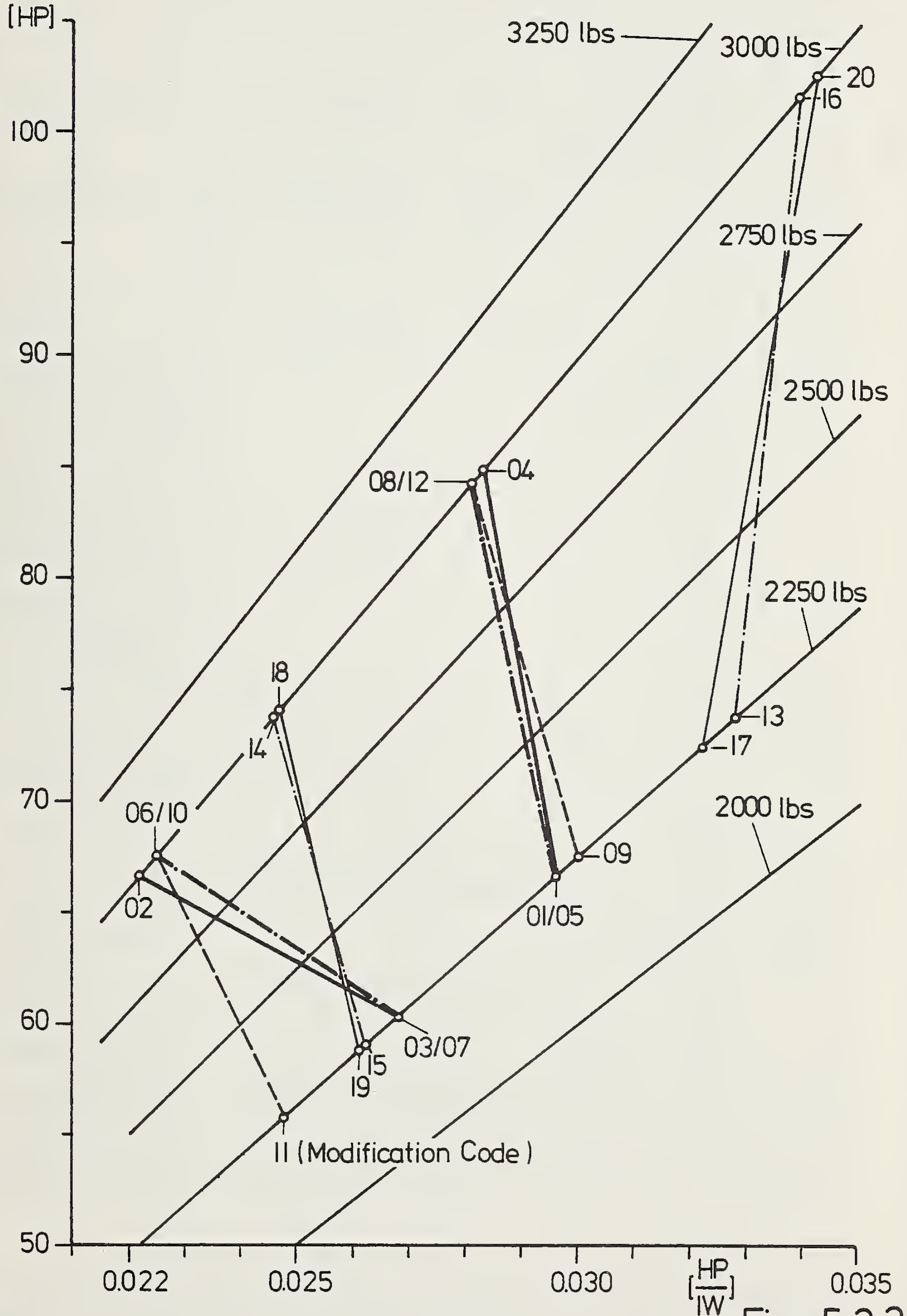

Engine Torque and Speed under Road Loading
[kpm]
15
10
5
0
2.2 l , 5 cyl. ~ 102 HP
1.6 l , 4 cyl. ~ 85 HP
1.6 l , 4 cyl. ~ 70 HP
1.3 l , 4 cyl. ~ 57 HP
50
100
150
[km/h]
200
20
30
40
50
60
70
80
90
[mi/h]
120
130
1000 2000 3000 4000 5000 6000 7000[RPM]
1000 2000 3000 4000 5000 6000 7000 [RPM]
1000 2000 3000 4000 5000 6000 7000[RPM]
1000 2000 3000 4000 5000 [RPM] 7000

Fig.: 5.2.1



[HP]
100
90
80
70
60
50
3250 lbs
3000 lbs
2750 lbs
2500 lbs
2250 lbs
2000 lbs
20
16
04
08/12
18
14
13
17
06/10
02
09
01/05
03/07
15
19
11 (Modification Code)
0.022
0.025
0.030
0.035
[HP/IW]

Fig.: 5.2.2

Vehicle No.	Modification Code	Type	License Plate	IW [lbs]	HP	$\frac{HP}{IW}$	F [m^2]	C_D	$F \times C_D$
1	01	Golf	WOB-VA 67	2250	66.7	0.0296	1.84	0.42	0.77
1	02	"	"	3000	"	0.0222	"	"	"
2	05	Rabbit	WOB-VD 93	2250	"	0.0296	"	0.43	0.79
2	06	"	"	3000	67.6	0.0225	"	"	"
2	09	"	"	2250	"	0.0300	"	"	"
2	10	"	"	3000	"	0.0225	"	"	"
3	13	"	WOB-VD 50	2250	73.8	0.0328	"	"	"
3	14	"	"	3000	"	0.0246	"	"	"
3	17	"	"	2250	72.5	0.0322	"	"	"
3	18	"	"	3000	74.1	0.0247	"	"	"
4	03/07	"	WOB-VX 36	2250	60.4	0.0268	"	"	"
4	11	"	"	"	55.8	0.0248	"	"	"
5	15	Scirocco	WOB-VS 35	"	58.9	0.0262	1.70	0.42	0.71
5	19	"	"	"	58.8	0.0261	"	"	"
6	04	Audi 100	WOB-VR 93	3000	84.9	0.0283	1.99	0.39	0.78
6	08/12	"	"	"	84.3	0.0281	"	"	"
7	16	Audi 5000	BS-JV 667	"	101.7	0.0339	2.0	0.42	0.84
7	20	"	"	"	102.6	0.0342	"	"	"

Tab.: 5. 2. 1

Modification Code	F [m²]	C_D	$F \times C_D$	Dyn. Wheel Diameter [cm]	Transmission Ratios Gears 1.	2.	3.	4.	Axle	No. of Cylinders	Displacement cu.in	Compression Ratio	Carburetor = C, Fuel Injection = FI	Air: Pump = P, Self Aspirating = S	EGR: On-Off = 0, Proportional = P	CAT: Ox. Cat = 0, 3-Way Cat = T	Closed Loop	Peak Horsepower	Max. Torque [kp×m]	IW [lbs]	HC	CO	NOx	UDC	HDC	Comb.
01	1.84	0.42	0.77	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	C					66.7	11.5	2 250	2.91	23.8	3.44	24.8	35.1	28.5
02	"	"	"	"	"	"	"	"	"	"	"	"	"					"	"	3 000	3.10	29.3	4.34	22.5	29.8	25.3
03	"	0.43	0.79	54.8	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"					60.4	9.4	2 250	2.44	26.5	2.02	26.9	40.9	31.8
04	1.99	0.39	0.78	60.0	"	1.94	1.29	0.91	4.44	"	96.9	"	"					84.9	12.6	3 000	3.13	27.9	3.74	21.1	31.3	24.8
05	1.84	0.43	0.79	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	C	S		0		66.7	11.5	2 250	0.59	7.26	2.18	23.9	35.1	27.9
06	"	"	"	"	"	"	"	"	"	"	"	"	"	"	0	"		67.6	11.9	3 000	0.71	9.69	2.00	21.7	30.5	25.0
07	"	"	"	54.8	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"	"		"		60.4	9.4	2 250	0.95	8.08	2.32	24.4	34.0	27.9
08	1.99	0.39	0.78	60.0	"	1.94	1.29	0.91	4.44	"	96.9	"	"	"	0	"		84.3	12.3	3 000	1.26	10.6	1.77	21.9	31.6	25.6
09	1.84	0.43	0.79	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	C	S	0	0		67.6	11.9	2 250	0.34	4.03	1.40	23.3	32.3	26.6
10	"	"	"	"	"	"	"	"	"	"	"	"	"	"	"	"		"	"	3000	0.40	6.50	1.76	22.0	30.3	25.1
11	"	"	"	"	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"	"	"	"		55.8	9.0	2 250	0.72	6.67	1.52	24.6	35.5	28.3
12	1.99	0.39	0.78	60.0	"	1.94	1.29	0.91	4.44	"	96.9	"	"	"	"	"		84.3	12.3	3000	0.62	7.63	1.74	20.9	30.8	24.4
13	1.84	0.43	0.79	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	FI	P		0+T	×	73.8	11.2	2 250	0.12	0.81	0.18	25.3	33.7	28.5
14	"	"	"	"	"	"	"	"	"	"	"	"	"	"	P	"	×	"	"	3000	0.16	1.24	0.18	23.5	30.6	26.2
15	1.70	0.42	0.71	54.8	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"			T	×	58.9	9.1	2 250	0.15	1.52	0.13	26.6	34.8	29.7
16	2.0	"	0.84	60.0	3.60	2.13	1.36	0.97	4.11	5	130.8	8.0	"	P	P	0+T	×	101.7	16.0	3000	0.10	1.30	0.18	16.3	24.2	19.1
17	1.84	0.43	0.79	55.8	3.46	1.94	1.29	0.97	3.90	4	96.9	8.0	FI	P		0+T	×	72.5	11.3	2 250	0.13	0.87	0.12	24.0	32.4	27.2
18	"	"	"	"	"	"	"	"	"	"	"	"	"	"	P	"	×	74.1	10.9	3000	0.16	1.51	0.13	20.5	29.1	23.6
19	1.70	0.42	0.71	54.8	"	2.05	1.35	0.96	4.06	"	77.6	8.2	"			T	×	58.8	8.9	2 250	0.23	1.66	0.10	24.6	36.5	28.8
20	2.0	"	0.84	60.0	3.60	2.13	1.36	0.97	4.11	5	130.8	8.0	"	P	P	0+T	×	102.6	16.0	3000	0.06	1.28	0.15	15.5	23.0	18.2

Tab.: 5. 2. 2

Engine Torque and Speed During Road Load									
		Vehicle No. 1/2/3 Modification Codes 01/02/05/06 09/10/13/14 17/18		Vehicle No. 4/5 Modification Codes 03/07/11/15 19		Vehicle No. 6 Modification Codes 04/08/12		Vehicle No. 7 Modification Codes 16/20	
		1.6 l ; ~70 HP		1.3 l ; ~57 HP		1.6 l ; ~85 HP		2.2 l ; ~102 HP	
km/h	mi/h	RPM	kpm	RPM	kpm	RPM	kpm	RPM	kpm
30	18.6	1070	1.58	1225	1.29	1070	1.86	1036	2.42
40	24.8	1430	1.79	1500	1.48	1430	2.08	1382	2.64
50	31.1	1790	2.01	1880	1.91	1780	2.36	1727	2.94
60	37.3	2150	2.36	2255	2.16	2140	2.72	2072	3.35
70	43.5	2510	2.72	2630	2.45	2500	3.15	2418	3.79
80	49.7	2870	3.15	3000	2.86	2850	3.65	2763	4.33
90	55.9	3230	3.65	3385	3.39	3210	4.15	3108	4.93
100	62.1	3590	4.15	3765	4.00	3570	4.80	3454	5.60
110	68.3	3950	4.80	4140	4.34	3920	5.44	3799	6.33
120	74.5	4300	5.44	4510	5.05	4280	6.16	4145	7.13
130	80.7	4660	6.16	4895	5.64	4640	7.02	4490	8.02
140	82.0	5020	6.95	5260	6.40	5000	7.88	4835	8.95
150	93.2	5390	7.81	5650	7.27	5350	8.81	5181	9.98
160	99.4	5740	8.67			5710	9.81	5526	11.07
170	105.6					6070	10.89	5871	12.23
180	111.8							6217	13.46
190	118.0							6562	14.76
200	124.2							6908	16.13

Tab.: 5.2.3

5.3 ELECTROMAGNETIC INTERFERENCE

The term 'electromagnetic interference' connotes a state of affairs where two pieces of electric or electronic equipment are functionally affected by mutual interference. To avoid malfunctions of this kind it is essential to know the source of the disturbance. It is for this reason that VW has developed measuring procedures and instruments which can be used to detect as well as to simulate disturbances of this kind.

There are two kinds of electromagnetic disturbance, interference by radiation and circuit interference.

Interference by radiation implies that the vehicle circuit itself is interfered with, which may occur either because of radio signals, such as those emanating from a radio station or a mobile transmitter (CB radio), or because powerful electric conductors are switched on or off. Generally speaking, only engines controlled by electronic devices are susceptible to disturbances of this kind. If, for instance, the electronic unit controlling fuel injection or ignition receives interference impulses there may be errors in the fuel amount injected or in ignition timing. This again may influence the fuel economy, the emissions, and the driveability of a vehicle in the manner described elsewhere.

Notwithstanding the fact that the metal body of a vehicle constitutes a welcome obstacle to interferences of this kind, VW has devoted much of their attention in the past to protecting the electric and electronic components especially of electronic fuel injection systems from interference by developing suitable shiedling equipment.

Circuit interference implies that the vehicle generates the disturbance and is affected by it at the same time. As far as electromagnetic interference is concerned, this aspect is more important than radiation.

Generally, disturbance inside the vehicle itself are not caused by the ignition system, but some powerful interference may be due to sudden load changes in the vehicle's electric supply system, i.e. to activating or deactivating powerful consumer units such as headlamps, or electric heating equipment. Extreme cases may even end in the mutual destruction of two units. It is, therefore, urgently necessary to avoid mutually detrimental influences of this kind in all vehicles.

Consequently, even test vehicles feature shielding elements in all pieces of equipment which might be affected by interference, and especially in the routing of wires great care is taken to ensure that interference is avoided. This process especially is governed by VW Procedures. As the requirements of these Procedures are generally met VW do not dispose of any quantitative data concerning possible connections between electromagnetic interference, fuel economy, emissions, and driveability.

Of course it is possible to combat electromagnetic circuit interference not only in a passive but also in an active way, namely by eliminating all sources of interference which might become active inside a vehicle wherever this can be done in the first place. Naturally, steps of this kind do not merely serve to cut down interference in pieces of electronic equipment required for the operation of the vehicle but also to suppress signals interfering with radio reception in the car.

Moreover, suppression of this nature also helps in eliminating electromagnetic impulses which might become interferences elsewhere. Vehicles are not allowed to emit any electromagnetic waves which might interfere with radio or TV reception in the houses or cars in the vicinity. Because of the high voltages involved, the ignition system is especially problematic in this aspect. As regards the frequency bands generally used for radio and TV transmission, i.e. from 20 to 1,000 megacycles, there is now a number of technical recommendations and even legal prescriptions which are followed by the industry.

The electric supply systems of the engine/vehicle systems analyzed in this Project are rather unsophisticated. The only piece of equipment whose scope exceeds that of conventional switches and ignition systems is the closed-loop control unit used for governing the air/fuel ratio in Modification Codes 13 - 20. These units use an input voltage of 0.5 to 1.0, which is equivalent to the voltage of the Lambda sensor, and an output voltage of 12 Volts because all they do is activate a simple check valve. Their design follows the shielding and suppression regulations of VW, which is why they did not cause any interference at any time.

6. RESULTS

6.1 FUEL ECONOMY

6.1.1 Methodology

There are two methods in VW use by means of which the fuel consumption of engine/vehicle systems is determined. The first method is the one described in the Federal Register, which involves testing complete vehicles on dynamometers according to the 1975 Federal Test Procedure (Urban Driving Cycle, UDC), measuring their exhaust emissions, and computing their fuel economy in mpg from these emissions. After this, the vehicle, which has warned up, is run through the Highway Driving Cycle (HDC), its fuel economy again being computed from its emissions. From the UDC and HDC fuel economy, the Combined fuel economy is computed as follows:

$$\text{FE (combined)} = \frac{1}{\dfrac{0.55}{\text{FE (UDC)}} + \dfrac{0.45}{\text{FE (HDC)}}} \qquad \text{I}$$

The second method of determining the fuel consumption of an engine/vehicle system involves testing the engine on an engine dynamometer controlled by the VW Programmed Control System (VW PCS). After having been brought to a temperature of + 20 °C, the engine is started cold, run through an UDC of 1372 seconds, run through another 1372-second UDC, warm start this time, and finally through a warm-start HDC.

Of course, the PCS must be fed beforehand with the data of the vehicle, especially the final drive ratio, the transmission ratios, the dynamic diameter of the tires, and finally the drag coefficient, the cross-sectional area, and the inertia weight.

While the engine is running, measurements controlled by the PCS are taken of the concentration of pollutants in the exhaust gas as well as of the quantity of air and fuel used. From those data, the emissions and, from the emissions, the fuel consumption is computed. Thus, three emission and three fuel economy ratings are computed for each pollutant from the UDC cold, UDC hot, and HDC tests. These emission and fuel economy figures are then categorized according to the accepted Federal Register procedure. We will now illustrate this by describing a sample fuel economy calculation.

To begin with, it should be noted that under the procedure described above the HDC test results do not have to be converted but are available for immediate use. The only operation still to be performed is merging these results and the results of the UDC test computations according to equation I into a Combined Fuel Economy.

In computing the UDC fuel economy the first consideration is that according to the Federal Test Procedure the results obtained from the first bag (505 seconds cold) determine the final result by a factor of 0.43, the results of the third bag (505 seconds hot) determine it by a factor of 0.57, and the results of the second bag, which is filled after stabilization, enters with a factor of 1. Thus, in accordance with the Federal Test Procedure, fuel economy is computed as in equation II:

$$FE\ (UDC) = \frac{0.43\ FEcd + 0.57\ FEht + FEs}{2} \qquad II$$

FEcd representing the fuel economy figure computed from the first bag,

FEht representing the fuel economy figure computed from the third bag, and,

FEs representing the fuel economy figure computed from the second bag.

As controlled by the VW PCS system, the UDC test is run through twice completely, a full 1372 seconds after cold and after warm start, thus producing two fuel economy figures. The cold-start fuel economy reading is weighted at 0.43, whereas the warm-start reading is weighted at 0.57, so that the overall UDC fuel economy can be computed after equation III:

$$FE\ (UDC)_{PCS} = \frac{0.43\ FE\ (\text{cold, 1372 sec.}) + 0.57\ FE\ (\text{hot, 1372 sec.})}{2} \qquad III$$

As it is permissible to assume that even after cold starting the engine temperature has stabilized after 505 seconds, it follows that equations II and III are completely identical, mathematically as well as physically, for equation III can be transcribed as follows:

$$FE\ (UDC)_{PCS} = \frac{0.43\ FE\ (\text{cold, 505 sec.}) + 0.43\ FE\ (\text{867 sec.})}{2} +$$

$$+ \frac{0.57\ FE\ (\text{hot, 505 sec.}) + 0.57\ FE\ (\text{867 sec.})}{2}$$

$$FE\ (UDC)_{PCS} = \frac{0.43\ FE\ (cold,\ 505\ sec.) + 0.57\ FE\ (hot,\ 505\ sec.)}{2} +$$

$$+ \frac{(0.43 + 0.57)\ FE\ 867\ sec.)}{2}$$

Therefore, fuel economy figures obtained on an engine dynamometer and on a vehicle dynamometer are bound to be the same. Results obtained from engine dynamometer tests offer an extra advantage in that they are more easily reproduced because on an engine dynamometer there are none of the individualistic influences exerted by human drivers on vehicle dynamometers (see also Chapter 3.2.3). Therefore, readings taken on an engine dynamometer are much less erratic.

Fig. 6.1.1 shows how closely an engine can follow a driving cycle in a PCS-controlled test. These graphs are test records made by the PCS system. The two lines in the bottom sector are records of velocity over time as prescribed by the Federal Test Procedure, one indication the prescribed velocity, the other indicating the actual velocity. The fact that the two lines are hardly distinguishable shows that in PCS-controlled tests the engine follows the prescribed speed curve quite closely.

As work on this Contract progressed, engines were subjected to engine dynamometer tests whenever there was some development work to be done, which usually happened with the low exhaust emission concepts. From the time when work on this Contract began to this very day, VW does and did not manufacture any engines capable of meeting exhaust emission standards as low as these. However, we already had the requisite knowhow and we had to bring the engines down to the levels required. This can be done much faster on an engine dynamometer than on a vehicle dynamometer, because all parts on which exhaust emissions depend are easily accessible, because, as can be seen from Fig. 6.1.1 results can be analyzed step by step, and because cold starts are possible at two-hour intervals, for both lubricant and coolant can be cooled down artificially. It is not necessary, therefore, to wait out the entire stabilization period of 12 hours normally required by an engine installed in a vehicle.

There is a third way of arriving at fuel economy and emission figures. However, the physical characteristics of this method are not equivalent to those of the other two. By this method, fuel economy and emissions are computed from engine maps either drawn up by PCS-controlled engine mapping or fed into the PCS system.

This method permits getting quick and reliable general information regarding the significance of, for instance, different transmission ratios, different inertia weights, air drag coefficients, installing different engines in the same vehicle, etc. In determining precise figures, however, this method must be used with caution because it will not account for certain factors which occur only in instationary operation. And, of course, it is impossible to obtain any cold-start results by this method. Still, the degree to which fuel economy figures computed by this method agree with fuel economy test results is astonishing. But what is really incredible about this procedure is its swiftness of operation. UDC or HDC warm-start test results can be produced after a mere 2 minutes! In Fig. 6.1.2 a few comparisons are made, the interrupted lines representing projections, the uninterrupted lines representing test results.

6.1.2 Emission Levels

In Figs. 6.1.3 through 6.1.6, average Combined Fuel Economy ratings are brought together with average emissions actually obtained from a number of engine/vehicle combinations meeting various emission levels (see Tables 6.1.1 through 6.1.5). Disregarding the difference between 4-cylinder and 5-cylinder engines, a certain trend is clearly distinguishable, namely:

The transition from uncontrolled to controlled concepts is accompanied by an improvement in fuel consumption. This tendency, however, remains unequivocal only in cases where the NOx standard is not below 1.0. If the NOx emission standard to be met is 0.4 g/m, which means that your engineering goal is 0.1 g/m (see also Chapter 6.2.3), you either have to use EGR or you have to shift your air/fuel ratio average further towards the rich side. This is done by setting the lambda control system so that the average air/fuel ratio is richer than it would be at the point where the conversion efficiency of CO/HC and NOx is the same (see Fig. 6.1.7). In the 1.3 l Rabbit, we were still able to use one catalyst and squeeze the HC and CO emissions in under the engineering goals in spite of the richness of the mixture for 0.1 gpm NOx, but in all other engines this was impossible. In all these cases, we had to use a second catalyst as well as a secondary-air system; in other words, we used a two-bed concept in spite of the lambda control. But, whatever solution may be ultimately adopted, fuel economy is bound to deteriorate in proportion.

Fuel economy either remains unaffected or deteriorates as a result of the transition from the old (3.1 g/m) to the new NOx emission standard (2.0 g/m). This is because in all concepts EGR must be used to meet the

present NOx standards, and because the mixture can be a bit richer if bigger catalysts are used. Surely this effect would have been enhanced if we had been trying to meet any engineering goals with Modifications Codes C1 through 12, but the tests on these Modification Codes had already been concluded when the demand for engineering goals came up, so that only the emission levels of 0.41/3.4/1.0 and 0.41/3.4/0.4 were affected.

The fuel consumption figures of the uncontrolled vehicles seem to be somewhat erratic compared to those of the controlled vehicles. The vehicles called uncontrolled in this context also have to meet emission standards, but these are European standards which are met by certain engine adjustments and manufacturing tolerances. What affects the fuel economy of these vehicles most seems to be the customer who eventually operates the vehicle. Thus, for instance, the uncontrolled vehicle whose inertia weight was large had a high fuel consumption, which suggests that everything here depends on driveability, whereas the fuel economy of the uncontrolled subcompacts is quite good, being surpassed only by that of the controlled vehicle operating at a NOx level of 1.0 g/m. The fuel economy of the small car fitted with the low-capacity engine was especially good because this particular engine is specifically designed for low fuel consumption.

In Fig. 6.1.8, UDC and HDC fuel economy figures are shown together with sulfate emission test (SET) fuel economy figures vs. various emission levels which had to be met (Table 6.1.6).

The vehicles used in these tests were not the ones used for the tests described in Figs. 6.1.3 through 6.1.6, but their equipment was the same. The tendency which appears here is the same we found before, the only difference being that the best fuel economy figures were obtained from Modification Code 13. This is because these vehicles were not checked continually during the sulfate tests, so that they did not meet the emission standards all the time, to say nothing of the engineering goals.

6.1.3 Vehicle Weight

Vehicle weight is one of the most significant factors influencing fuel economy. On the road, the total weight of a vehicle must be accelerated and decelerated again and again. Each acceleration consumes a certain amount of energy which is not recovered during deceleration. The larger the weight of a vehicle, the larger the amount of energy required to accelerate it and, therefore, the loss of energy during deceleration and, finally, the amount of fuel consumed. It is obvious that the effect of this influence will decrease as the movement of general traffic grows more uniform.

The vehicle rate is composed of its inertia weight and its payload. The payload influence is discussed in Chapter 3.2.2, therefore we should concentrate our considerations here on inertia weight.

All essential results of the tests run under the provisions of this Contract are shown in Fig. 6.1.9 through 6.1.14 and Tables 6.1.1 through 6.1.5. It is discernible that the influence of the inertia weight on fuel economy as determined by UDC testing is more or less constant up to emission level 0.9/9.0/2.0, whereas in vehicles with a low horsepower-to-inertia-weight ratio (HP/IW) the influence of the inertia weight was more noticeable in HDC tests. The reason is that in HDC tests relatively weak engines, especially if they have to move a large mass, get into the full-throttle range again and again. The fact that the picture presented by emission levels 0.41/3.4/1.0 and 0.41/3.4/0.4 is different, especially whenever larger masses are concerned, is due to the less economical 2.2 l 5-cylinder engine being used in these cases.

It may be said that, generally speaking, the influence of inertia weight is quite uniform. Fig. 6.1.14 even indicates that its influence seems to be linear, at least in the 2000 to 3000 lbs weight range. There is a comparison of various results obtained from the 1.3 l engine, namely 2 cold-start UDC tests run on an engine dynamometer, 3 UDC and 3 HDC warm-start engine dynamometer tests (thin uninterrupted lines), and fuel economy projections from engine maps for inertia weights of 2000, 2250, 2500, 2750, and 3000 lbs. Any differences between the results of cold and warm-start tests are due to cold-start enrichment, whereas those between warm-start and projection figures are due to acceleration enrichment which, in this lean engine, leads to an improvement in fuel consumption.

In addition to these data, we used others obtained by fuel economy warm-start tests run on production vehicles which were performed on vehicle dynamometers, (Fig. 6.1.15 (1)). The lines in this figure indicate fuel consumption expressed in g/mi at inertia weights from 1750 to 2750 lbs. The circles represent the mean of the readings, whereas the vertical lines projecting from them upwards and downwards represent the scatter bandwiths. This figure indicates that there is a good correlation between inertia weight and fuel consumption, the latter increasing by 8 % for each additional 1000 lbs of inertia weight. The results here are on the same scale as those obtained during the tests performed under this Contract, but they are distinctly lower in absolute figures, the increase here amounting to about 10 - 11 % for 1000 lbs.

In spite of what Fig. 6.1.14 indicates there can be no linear correlation between inertia weight and fuel consumption because the influence of the mass of the vehicle increasing or decreasing by one unit must necessarily be less in larger vehicles. This is confirmed by Fig. 6.1.16,

which shows data obtained during EPA certification tests on vehicles equipped with spark ignition engines made by various manufacturers, plotted over inertia weight. It can be seen that, quite according to expectations, the influence of inertia weight decreases as the mass itself increases. In this process, however, the dispersion of individual results increases as well.

There is a certain rough and ready formula describing the influence of inertia weight on combined fuel economy in smaller vehicles which we investigated: 2 % per 100 lbs. Fig. 6.1.16 shows a relation 3.5 % per 100 lbs.

The inertia weight of a vehicle can be modified, though not to the extent that Automobile Manufacturers can choose the inertia weight of a vehicle at will. The lowest possible inertia weight of a vehicle is determined by the following parameters:

- Passenger safety
- Operational safety
- Comfort

The more stress is laid on these parameters, the higher the inertia weight of the vehicle will be at a given technology of body design, which means naturally a corresponding loss of fuel economy. For this reason, it is necessary to work out sensible compromise solutions as far as these aspects are concerned.

In addition to this it is possible to thoroughly check commercial vehicles for ways in which their inertia weight may be reduced without loss of either safety or comfort. It is also possible to consider new materials which are lighter but offer the same strength. The final alternative is to reduce the size of all vehicles as it is extremely rare to have more than two persons traveling in the same vehicle. To transport two persons, however, passenger compartments need be much less roomy than those of today's cars.

6.1.4 Engine Peak Horsepower to Inertia Weight Ratio

In Figs. 6.1.9 through 6.1.13, which we have already seen in Chapter 6.1.3, all points on the graphs are accompanied by their HP/IW ratio. Barring a few exceptions, the fuel economy of vehicles having low HP/IW ratios is superior. However, this should not mislead anybody into concluding that the relationship between fuel economy and HP/IW ratio is unambiguous.

A study which was made for the Government of the Federal Republic of Germany and which involved fuel economy tests run according to the U.S. Test Procedure, showed that there is no correlation between HP/IW ratio and fuel consumption. This is corroborated by Fig. 6.1.17, which shows the fuel consumption of a number of uncontrolled vehicles manufactured by VW. The first thing which strikes you here is the ruling influence of inertia weight. Test results obtained from vehicles of the same inertia weight from curves which droop as the HP/IW ratio increases. It is precisely this aspect, i.e. fuel economy deteriorations corresponding to increases in the HP/IW ratio, which is responsible for the seeming interdependence between fuel consumption and HP/IW ratio, which is evident in Figs. 6.1.9 through 6.1.13 That there is no such interdependence is demonstrated by Fig. 6.1.17, for it is possible for fuel consumption figures from 25 to 42 mpg to be found in connection with HP/IW ratios anywhere between 0.02 and 0.04.

As the same figure shows, peak horsepower is another characteristic of major importance, because it can be seen that fuel economy deteriorates as the peak horsepower increases. Naturally, the inclination of these lines is such that increasing the inertia weight with the peak horsepower remaining constant means that the fuel consumption will increase as well. This tendency intensifies as the HP/IW ratio decreases, because the lower the peak horsepower, the more fuel economy will be affected by additional mass.

Fig. 6.1.18 explains why fuel economy depends so much on peak horsepower. This figure shows the engine maps of a high-horsepower and of a low-horsepower engine, indicating that, provided that the transmission and final drive ratios are properly designed, the engine whose peak horsepower is higher will always run at comparatively low speeds. Thus, for instance, road load point A of the weaker engine corresponds to a certain engine speed n_1, which is higher than n_2, which corresponds to the stronger engine's road load point B. This would seem to warrant the conclusion that the more powerful engine combines less extensive friction losses and a better fuel consumption. This is not so, however. Fig. 6.1.19 demonstrates this by a comparison of the friction and ventilation losses exerted by the two engines at speeds n_1 and n_2 (see Table 6.1.8). At the lower speed, the losses of the larger engine is higher by 17 %. But it is not merely a question of engine speeds alone; another factor is that of point B being situated much lower on its map than point A, meaning that A is much closer to the area of minimum fuel consumption, which is represented by A_0, than B is to its point of minimum fuel consumption, B_0. Briefly, the specific fuel consumption of A is 310 g/kWh, whereas that of B is 350 g/kWh.

This is because there is more friction in the larger engine, and also because the thermodynamic conditions under which the more powerful engine operates at point B are worse than those under which the less powerful engine operates at point A.

To make this point quite clear we have shown in Fig. 6.1.20 nothing but the relationship between fuel economy and peak horsepower, the parameter being inertia weight. One is instantly reminded of Fig. 6.1.17, but it is clear now that the seeming relationship between fuel economy and HP/IW ratio is really one between fuel economy, IW, and peak horsepower.

6.1.5 Transmission and Drivetrain

Fig. 6.1.21 and Tab. 6.1.9 show the influence of the final drive ratio on fuel economy. The unbroken lines connect hot-start test results, whereas the interrupted lines represent projections, i.e. fuel economy figures computed from engine maps drawn from UDC and HDC tests. The reason why the projected fuel economy figures are lower is the same as that given with Fig. 6.1.14.

It can be seen that varying the final drive ratio will affect HDC tests slightly more than UDC tests, and that within the field which is being discussed here there is a near-linear relationship between both fuel economy readings and the final drive ratio, which of course means that fuel consumption increases if the engine speed goes up.

The degree of influence exerted by the diameter of the wheels is the same as that of the final drive ratio.

Furthermore, we studied the influence of using various transmission ratios. Figs. 6.1.22 and 6.1.23 with the Tab. 6.1.10 show a comparison by projection between our elected drivetrains and the 4-speed standard drivetrain. On the positive vertical axis, we have UDC and HDC fuel consumption figures. On the negative vertical axis, we have the product of final drive ratio and gear ratio, i.e. the ratio between the engine speed and the speed of the driven wheel in each gear. The base blocks show the final drive ratio and the small blocks below pertain to each gear.

It can be seen that a 5-speed gearbox designed for sport cars produces fuel economy results which are nearly identical with those of the standard 4-speed transmission, the only difference being a 0 to 2 % worse fuel economy in the UDC, because in 3rd gear the speed ratio between engine and driven wheel is somewhat higher. To compensate for that, the 5-speed drivetrain performs ~ 2 % better in the HDC, because in 5th gear the speed ratio is somewhat lower than that of the 4-speed gearbox in 4th gear.

Keeping in the first four speeds the same transmission ratios and the axle ratio as before but designing the fifth speed as a genuine overdrive with 22.6 % less engine speed produces the same fuel economy in the UDC, because the 5the gear is not used at all, practically speaking, and a significant improvement of ~10 % in the HDC.

If instead of the standard 4-speed drivetrain you install a 4-speed drivetrain in which the transmission ratios of speeds 1, 2 and 3 are the same as in the standard version but the 4th speed is an overdrive gear, the fuel economy results obtained will again be nearly the same as those obtained from the 5-speed drivetrain with the 5th gear acting as overdrive. Only the UDC results are ~1 % better. This is because in the 4-speed drivetrain the transmission ratio between engine and driven wheel is somewhat lower in 3rd gear than in the 5-speed overdrive gearbox.

Another great step towards better fuel economy can be made by installing a 5-speed drivetrain which uses the lowest final drive ratio which is feasible in VW drivetrains. Speeds 1 through 4 of that drivetrain, have the same transmission ratio as the standard 4-speed drivetrain, but the number of gears in 5th gear is the same as that of the overdrive used in the 5-speed and 4-speed overdrive gearboxes. A drivetrain of this kind will make possible fuel consumptions of more than 32 miles/gallon, i.e. 9 % better in the UDC and of nearly 40 miles/gallon i.e. nearly 15 % better, in the HDC, but the driveability of such a system would be unsatisfactory.

To corroborate our conclusions regarding the influence of changing the transmission ratio of individual gears we again modified the third gear of the five-speed transmission. The results are shown in Fig. 6.1.23.

The speed ratio between engine and driven wheel in 3rd speed was reduced slightly. We found that the HDC fuel economy is the same as that of the sports 5-speed drivetrain, but that there is a gain of a scan mile/gallon in the UDC.

Fig. 6.1.24 shows the UDC, HDC, and Combined fuel economy figures of a number of engine families manufactured by VW and certified for 1977. The vehicles are equipped with manual as well as with automatic transmissions. The graph shows that automatic vehicles use between 10 and 15 % more fuel under highway driving conditions than vehicles with manual transmissions.

When running under urban driving conditions, however, automatic vehicles do not consume any more fuel than vehicles equipped with manual transmission, which makes for a corresponding dibution of the differences between automatic and manual vehicles in the Combined fuel economy figures.

We may reasonably expect the fact that automatic transmissions hardly influence the Urban fuel economy at all to be due to the strict emission standards governing city traffic. All ranges on the engine map which come into play during urban driving are kept as lean as possible to keep the emission of pollutants down. It is because of driveability factors that engines linked to automatic transmissions can be operated on leaner mixtures than engines on manual. Any faulty driveability of an engine will be damped by the automatic transmission so that it shows much later than in a manual drivetrain.

For illustration purposes, Figs. 6.1.25 and 6.1.26 show again fuel economy figures of a Rabbit and a Dasher with manual and automatic transmission under cruise conditions. Here again, there is a difference at high speeds amounting to 10 to 15 %, which decreases with the speed.

6.1.6 Air Drag

The engine power required to maintain a constant speed on a level road under conditions of no wind is made up, as is well known, of the power needed to overcome internal losses and the power needed to overcome external resistance:

$$P = \frac{1}{\eta_G} \times (P_D + P_R)$$

where P_D and P_R are the air drag and rolling resistance power requirement respectively, and η_G is the efficiency of the drivetrain.

The relationship between air drag and total external resistance to motion of a vehicle is shown in Fig. 6.1.27 (2) and 6.1.28 (3). Above a speed as low as 40 mph with a light car the proportion represented by air drag amounts to more than 50 percent of the total drag. With a heavier car this point will be reached at 60 mph. Depending on the drag coefficient and the weight, the air drag amounts to 80 or 90 percent of total drag at higher speeds.

The momentary fuel consumption is determined from the required engine power and the engine's specific fuel consumption:

$$B = P \times b$$

with fuel consumption B (g/h), engine power P (kW) and specific fuel consumption b (g/kWh).

This specific fuel consumption can be taken from the engine map. If the air drag of a vehicle is lowered without alteration of the drivetrain and the engine, the load curve shifts towards a lower mean effective pressure at the same engine speed. The new load point is associated with higher specific fuel consumption (Fig. 6.1.29 (3) broken and dotted line).

If the new load curve is extended to the intersection with the engine's full power curve, we obtain a new maximum speed point. If the reduction in air drag is sufficient, therefore, the maximum permissible engine speed will be exceeded.

As the power requirements are reduced, a drop in fuel consumption is achieved; the effect of this is however diminished by the increase in specific fuel consumption.

In order to make use of the full potential of reducing the air drag, the transmission ratio must be adjusted, as shown in Fig. 6.1.29. The load points will move along hyperbolas of constant power into a region of lower specific fuel consumption.

By means of such a transmission ratio adjustment, the new load curve can be brought more or less into alignment with that of the vehicle in its original state, though in this case the same engine speed, point (1), will be equivalent to a higher road speed.

The increase in top speed produced by lowering the drag coefficient with and without gear ratio modification can be seen in the lower part of Fig. 6.1.29.

Thus, reducing the air drag is especially helpful in lowering fuel consumption and increasing the performance of a vehicle. The improvements in each case are governed by the following rule-of thumb formula:

$$\text{Air drag reduction by } 10\,\% \begin{cases} 3.5\,\% \text{ improvement in Combined fuel economy} \\ 2.5\,\% \text{ improvement in Urban fuel economy} \\ 5\,\% \text{ improvement in Highway fuel economy} \\ 3\,\% \text{ improvement in top speed} \end{cases}$$

Fig. 6.1.30 defines the term 'air drag' as a combination of the air drag coefficient (c_D), of the frontal area of the vehicle (A), and of the pressure of the air flow towards the vehicle.

C_D is a characteristic describing the aerodynamic quality of the shape of a vehicle. In a full size wind tunnel, c_D can be measured very precisely, very quickly, and at a rather early stage in development, contrary to c_D measurements on the road. In a wind tunnel, c_D changes of 1 % can be measured. The P_D equation is shown here to recall the fact that the engine power required to overcome air resistance increases as the third power of the vehicle's speed, so it grows to eight times its original value as the vehicle speed doubles.

For design engineers, there is little room to maneouver as far as the size of the vehicle as expressed by its projected area lengthwise is concerned - $\pm$ 2 % is the limit - but the c_D quotient still has a considerable development potential.

The bar diagram in Fig. 6.1.31 gives an overview of the c_D quotients of modern European automobiles. The mean from all 91 standard automobiles analyzed is $c_D = 0.46$. The difference between the vehicles most and least sophisticated aerodynamically amounts to 30 %.

Fig. 6.1.32 (3) represents an attempt to illustrate the trend in the development of automobile air drag. Reliable data are available only for vehicles of fairly recent date. A statistical evaluation of European mass-produced vehicles has been undertaken; but there is no breakdown into years of manufacture. Comprehensive data are required to draw up an accurate picture of the development of drag coefficient. Such data, however, are not available. Therefore, we used results published for some older vehicle shapes and certain measurements made of scaled models some time ago (3).

The drop in drag coefficient from $c_D = 0.8$ for the cars of the Twenties to an average of $c_D = 0.46$ for modern European automobiles has occured in two stages. In the first stage (between the two world wars) cars grew longer, lower, and smoother in detail while still retaining such distinguishing features as non-integrated fenders and headlights. They reached approx. $c_D = 0.55$.

The second stage in the reduction of air drag was the introduction of cars having fully integrated bodies with notch-back, fastback, or squareback. The incorporation of fenders and headlights into an all-enveloping body greatly improved the airflow around the vehicle. Depending on detail design, drag coefficients of $0.4 < c_D < 0.5$ were now attained. This has not altered since about 1960. Admittedly, special body shapes have been designed to achieve lower c_D values; but in the bulk of European automobiles in the past 15 years no tendency of a systematic reduction in drag coefficients can be detected.

The drag coefficient of $c_D = 0.15$, which in Fig. 6.1.32 is given particular prominence, was attained as early as 1922 by the vehicle shema at the bottom left side which was designed by W. Klemperer. Shown other shapes, for example that on the right, have been used by Volkswagenwerk recently to obtain equally good c_D values. This means that $c_D = 0.15$ can be regarded as the probable optimum even if this figure is not based on any form of natural law. The gap between this value and those achieved by modern mass-produced automobiles is considerable.

Fig. 6.1.33 (3) shows fuel economy versus speed for different c_D values. It refers to steady state conditions.

But in order to get a realistic picture of the advantage of air drag reduction from the average driver's point of view, fuel economy of three different standard cycles was calculated taking into account the transients. The results are shown in Fig. 6.1.34 (3). As expected, the gain in fuel economy due to air drag reduction is most striking in the highway cycle. It is still noticeable in the composite cycle, which reflects day-to-day driving conditions in the US most closely.

To achieve drag coefficients 10 to 15 % better than the European average VW developed their so-called optimizing method.

It is based on the postulate that the styling concept of a vehicle must be accepted as it stands. Aerodynamic improvements can only be attempted in the form of detail changes. These must be executed without changing the vehicle's appearance.

Front End Design

An example for this procedure, taking into account the peripheral conditions referred to, is shown in Fig. 6.1.35 which summarizes the main results of shape optimization on our new Dasher Design.

As can be seen from design A the air flow separated slightly from the engine hood of the Dasher models manufactured before 1977. Because of this, the air drag coefficient of this vehicle was not better than $c_D = 0.46$ in spite of the fastback design, a figure which corresponds exactly to the average air drag coefficient of all European automobiles.

By attaching a nose B whose design was governed by aerodynamic considerations only we intended to demonstrate the extent to which a reduction of the air drag coefficient is possible without abandoning any of the major dimensions of the vehicle front. The figure shows that an optimum nose and spoiler will reduce air drag up to 15 %. This figure constitutes the limit of what is attainable by means of feasible design alterations.

Changing the shape of the nose as under D, i.e. by attaching an aerodynamic molding, will reduce c_D by a mere 2 %. Adding a front spoiler of optimum shape will also reduce c_D by no more than 2 % if there are - no other changes. This reduction, however, is accompanied by a remarkable drop in the front lift coefficient C_{LF} which amounts to as much as 30 %. Even by combining C and D you get a c_D reduction of a mere 7 %, which is still considerably less than that produced by the optimum front end design B, namely 15 %.

Only by redesigning the entire front end of the vehicle including the bumper did we succeed in E a design whose c_D quotient is 0.40, quite close to the optimum of 0.39. In addition to an air drag reduced by 13 %, the nose of the '78 Dasher model also produces a front axle lift which is not more than half of that of the former model. A lift as low as this will not only improve driveability but also prevent the body of the car from lifting at high speeds and induce less bending in the drive joints, thus making for a reduction of wear.

When designing the front end of the new Dasher model we were aided by the new bumper concept. As bumpers now rely on concealed steel bars for strength, we were able to design their visible plastic surfaces so that air drag was kept low and the air was channeled properly for cooling.

As is indicated by bar E the c_D coefficient of the new Dasher will vary from 0.40 to 0.41, depending on the kind of engine and headlight equipment used.

Fig. 6.1.36, which is a photograph taken in VW's wind tunnel, shows the flow of air around the '78 Dasher. The air now follows the contour of the nose without separation and it continues to follow the contour of the car towards the rear until it separates at the rear edge of the trunk, where the wake is clearly visible.

In the '78 Dasher, the c_D improvement has led to a reduction in fuel consumption amounting to 0.08 to 0.16 gal/100 mi, depending on the engine version and the velocity, while the top speed of the vehicle is increased by about 3 mi/h without any increase in engine power output.

In another investigation the front end of a fastback sedan was optimized. On account of the complex rear end flow pattern, which is three-dimensional in nature, this type of tail reacts quite sensitively to any change in the flow at the tail end of the vehicle.

When optimizing front spoilers in the wind tunnel, the quality of the airstream close to the ground and the ground boundary layer thickness are of particular importance. The displacement thickness of the ground boundary layer in the VW climatic wind tunnel is only some 10 % of an automobile's ground clearance, so that even spoiler tests can be carried out unrestrictedly, as comparative road tests have shown.

The effect of front spoilers on drag and lift for a coupê is shown in Fig. 6.1.37 (3).

Tests have been on inclined and vertical spoilers the latter being mounted at three different positions on the front of the car. No definite influence on detected, but there was a considerable decrease in lift, corresponding to the length of the spoiler. Spoiler B40 yielded the lowest air drag, the reduction in drag coefficient c_D being 3 % compared to the car without spoiler; the front axle lift c_{LF} was reduced by 21 %. A short spoiler (pattern B40) was recommended as a simple method of reducing drag. Up to 15 % drag reduction has been achieved, depending on the car under consideration.

The influence of leading edges on drag is becoming more important, since, on contemporary passenger cars, the general outline of the vehicle front is approximately fixed by the position of the engine, the position of the front bumpers, headlights, and traffic lamps to meet the demands of the law.

Fig. 6.1.38 (3) shows the optimization of the vehicles' leading edges. By mounting the "optimum nose", consisting of the two parts M1 and K1, a decrease in drag of Δc_D = 0.05 was achieved. Rounding off the vertical leading edges only by means of the ancillary front and section, K1, decreased the drag coefficient by Δc_D = 0.015, i.e. by only 30 % of the full drag reduction achieved by the complete optimum nose. The ratio of drag reduction of 70 % and 30 % of both well-rounded hood and vertical leading edges of the fenders is, in general, certainly not transferable. This ratio is influenced by the lateral curvature of the vehicle's front end, position and shape of the front bumper, flow pattern in front of the radiator, angle of incidence, ground clearance of the vehicle and inclination of the front hood.

The optimum nose was removed and by a step by step rounding off of both horizontal (m) and vertical (k) edges, an attempt was made to duplicate the results obtained with the optimum nose. Finally with the combination M3 and K3, a drag reduction of Δc_D = 0.045 was achieved. This figure, 90 % of maximum drag reduction, was achieved with a contour still acceptable to the stylist.

On vehicles with a short engine hood length, the front edge not only influences the drag but also the pressure at the area in front of the windshield, which in turn governs fresh air entry for heating and ventilation, (see Fig. 6.1.39 (3)). Version A with fully-separated hood airflow - caused by the sharp-edged front end contour - produces a high drag coefficient of c_D = 0,48. Since the separated flow does not reattach on account of the short hood line, only a slight pressure of c_p = + 0.10 obtains at the base of the windshield. This pressure c_p is defined as $c_p = \Delta p/q_\infty$ where Δp is contour pressure relative to atmosphere and q_∞ is dynamic airflow pressure. If contour A were to be used, the low pressure at the inlet grille would mean that the fan would have to run continuously.

In the case of contour B, on which the flow around the front edge was improved by an aerodynamically sound molding, the airflow separates from the surface but reattaches at the area of the inlet grille. On this version, the static pressure at the inlet grille increases to $c_p = + 0.30$. The drag coefficient is reduced by the front edge molding slightly.

A rounded-off front edge as in contour C yields a flow over the hood free from separation, associated with a drag 15 % lower than that of the sharpedged front end from A, namely $c_D = 0.41$. The pressure on the inlet grille in this case is $c_p = 0.40$, thus ensuring a high fresh-air volume without running the blower.

Design of the A-pillar

The influence of the design of the front roof support (usually called the "A-pillar") on aerodynamic drag is dependent to a large extent on position and curvature of the windshield and on the design of the vehicle's front. Attention has to be paid to these parts of the body because of manufacturing requirements, the obscuring of side windows by water and dirt, wind noise, and pressure conditions at the door gaps.

Fig. 6.1.40 3 shows the influence of the A-pillar design on drag in a coupé. The original design 1 with a prominent rain gutter (drip molding) represents a convenient solution from the manufacturing point of view. Separation of the airflow immediately behind the A-pillar, however, is responsible for an undue drag increase and loud wind noise.

The flow of water as illustrated in Fig. 6.1.40 is stopped by the rain gutter, thus preventing the side windows from becoming soiled (if the rain gutter is properly designed). If the lateral curvature of the windshield is small, the water will overflow the rain gutter and spread over the side windows. Under conditions of heavy flow separation, a fine water spray is created, distributing itself over the outside mirror and side windows. This will affect visibility - especially in the outside mirror - and endanger safety.

A-pillar 2 without the rain gutter yielded a drag reduction of 7 %. Wind noise was also reduced due to a considerable prevention of flow separation. A disadvantage of A-pillar 2 is the fact that rainwater can flow over it undisturbed, thus soiling the side windows.

The A-pillar design 3 shows a rain gutter with its flange mounted partially flush with the outer body of the pillar. This design, in comparison to 1, produced a drag reduction of 5 % and a 3 % increase compared with design 2, which has no rain gutter. Soiling of the side windows did not occur in design 3.

Design 4 shows a rain gutter fully incorporated into the A-pillar, thus giving the same drag coefficient as form 2, which has no rain gutter. The latter design has the advantage of presenting less difficulty in manufacturing and keeping the side windows free of water. No difference in wind noise was found between design 2 without a rain gutter and design 4 with the flange incorporated.

A-pillar 5 is characterized by a long watercatching "pocket" and a side window nearly flush with the outer body. The drag coefficient is 3 % less than on design 2, which has a prominent flange. As the side windows are nearly flush with the outer body, the airflow around the A-pillar is nearly undisturbed, resulting in low wind noise.

The extent to which the drag is affected even by quite small changes of the A-pillar, is shown by the examples in Fig. 6.1.41 (3). The basic design A has no specific features intended to keep the side windows clean. The pillar is aerodynamically good, the airflow over it being almost free from separation effects. In version B, a contour change was attempted on the outer face of the A-pillar, in order to prevent rainwater from dropping into the interior of the car when it is stopped and the door is opened. On this version of A-pillar B, the water flows from the roof gutter into the recess in the pillar contour and downwards. The recess does not adversely affect the drag coefficient.

In the case of design modification C, the same water drain effect at a standstill has been achieved as on version B. The drain lip here, however, is an outward extension of the pillar contour, as shown in the sketch. This leads to an airflow separation at the pillar, which in turn leads to a 4 % increase in drag.

Whereas pillars B and C only ensure reliable shedding of rainwater when the car is at a standstill, version D is also capable of keeping the side window clear when on the move. The water flowing over the windshield is collected in the drain gutter and is conducted down and also into the roof rain gutter. In this design, where the window is fixed by "glueing", the drain channel is formed by the pillar contour and by the shaped trim surrounding the windshield. In order to ensure that this integrated rain channel operates reliably, it is necessary to optimize it for each new vehicle, since the styling of the windshield, its inclination and the contour of the A-pillar influence the shape of the channel. The drain channel shown as D keeps the side windows entirely clear of water and dirt at all speeds even if a side wind is blowing and it does not lead to any increase in drag coefficient.

Rear end design

Whereas in the case of front end design the results can be used with only minor limitations for other vehicles of different types, provided that the principal dimensions are similar, any such duplication of optimizing results at the rear end of the car is scarcely possible. The flow pattern at the rear of the vehicle is determined by the flow regime of the front and of the windshield, and also by the overall dimensions. Furthermore, the three-dimensional nature of the rear end airflow is influenced by the details of the rear end design.

As an example of rear end optimization on a notchback car, Fig. 6.1.42 (3) shows the influence of the height of the trunk lid on the drag coefficient c_D. The graph shows that a slight lowering ($z = -50$ mm) or raising to a height of $z = 100$ mm does not bring any change in drag.

With rear end elevations between 100 and 150 mm, the drag coefficient drops from the basic shape, with $c_D = 0.40$, by 8 % to $c_D = 0.37$. The contour of this form A with 150 mm lid elevation can be seen in the sketch.

Still further elevation of the rear end brings us into the realm of the station wagon as shown by the contour referred to here as form B. This yielded a drag coefficient of $c_D = 0.38$; in other words, 3 % above the optimized notchback form A but 5 % below the original basic form.

The tests on the influence of trunk lid elevation were conducted without changes to the side panels, so that airflow onto the tail remained unchanged. The influence of narrowing the rear and side panels is shown for the same vehicle as before in Fig. 6.1.43 (3). The rear end side panel contraction y was investigated in stages starting with straight side panels as the basic form. The trunk lid height remained unchanged as the side walls were made narrower. In the region up to y = 50 mm (design A), a continuous drop in drag was established until the improvement amounted to 5 %, whereas the specific contraction range between 75 and 125 mm brought with it a sudden drop from $c_D = 0.43$ to $c_D = 0.37$, a total of 13 % (design B). As the side panel was narrowed down to y = 200 mm on each side of the vehicle (design C), the drag remained unchanged.

The following remarks deal with the influence of the rear end inclination angle of fastback and squareback vehicles. A fastback rear end is defined as one on which the separation line is located at the base of the sloping rear panel. The rear window, which forms part of the sloping panel, is in a region of attached airflow and thus remains free from dirt. The square end body, on the other hand, possesses a separation at roof level. The entire rear end including the rear window lies in the separated airstream so that dirt is deposited more or less on the glass in the car's wake, depending on the body's rear overhang and the flow pattern in the wake.

The influence of the rear end angle of inclination on drag and the location of separation can be seen in Fig. 6.1.44 (3). In vehicles with a steep angle tail panel, for instance station wagons with $\varphi > 35°$, the point of separation is at the rear edge of the roof. In the example shown, the drag coefficient is relatively low at $c_D = 0.40$. If φ is reduced to a smaller value, the separation line moves from the rear edge of the roof down to the lower edge of the inclined rear panel. The drag is increased along with the downshift of the separation line; in the example illustrated, the increase is 10 %, to $c_D = 0.44$. The higher drag is attributable to strong trailing vortices with a corresponding rise in lift and thus in drag.

The transition from square end tail to fastback does not take place suddenly at a specific inclination angle limit, but in a transitional zone shown as a shaded area on the graph. In this transitional zone, the separation line oscillates between top and bottom, the degree of fluctuation depending on the edge pattern and the speed. If the angle φ is reduced still further, the drag drops again. At a fastback inclination angle of $\varphi = 23°$ the same drag coefficient of $c_D = 0.40$ is obtained as for the squareback flow pattern.

This angle $\varphi = 23°$ represents the approximate limit of what is acceptable in a sedan, allowing for a reasonable angle of rearward vision. Smaller angles of approx. 15°, have been applied in coupés with c_D values up to 15 % lower than those obtained with a squareback.

The relationship between drag and rear end inclination referred to here applies to medium-sized automobiles with an attached upstream flow field. If the flow pattern at the front section of the car is unfavourable the limit angle is reduced from 30° to approx. 25°; if the front has an excellent flow pattern and no separation occurs at the windshield the limit angle can be 35°.

The influence of the rear spoiler on drag and on the rear axle lift in coupés is shown in Fig. 6.1.45 (3). The separation line of all shapes shown is located on the lower rear edge of the body. Modifications 2 to 6 yielded different improvements in drag and lift. Spoiler 5 finally was adapted for this car.

Small changes in the contour of the rear end of the roof as well as in the C-pillar may lead to a considerable drag reduction. Fig. 6.1.46 (3) compiles some specific results achieved with a squareback car. The rear end of the roof was rounded off as shown in detail D, leading to a decrease in drag of 9 %. Building up the C-pillar as shown in section C - C yielded a lower drag too. But what surprised most of all was the fact that none of the combinations of roof and side panel modifications led to a drag improvement of more than 9 %.

From Fig 6.1.47 (3) it can be seen that the drag is not only influenced by the outer contour of the body but also by small gaps of lids. The example shows the effect of the rear door lid gap on the wake and thus on drag.

If the gaps between the rear door and the roof and the C-pillar are open, a low drag coefficient of $c_D = 0.42$ is obtained together with an airflow separation line at the rear end of the roof. If the two side gaps are sealed there is neither a change in drag nor a change in the wake pattern. However, if the roof gap or both roof and side gaps are sealed, the separation line is shifted down to the lower edge of the rear door. The strong trailing vortices associated with this condition cause high induced drag which shows up as a 14 % increase, $c_D = 0.48$. At the same time the lift coefficient c_L rises by some 28 %.

Fig. 6.1.48 (3) shows streamlines on a VW SCIROCCO which was optimized by the method just described. It is clear that there is no separation of the airflow over the entire front part of the car. The deflecting action of the integrated front spoiler is obvious. The point of separation at the rear is not clearly visible. By filling up the separated flow of the wake with smoke, it can be shown that the flow separates at the rear spoiler. The picture was taken with the aid of multiple exposures of a single smoke trail, the height of which was varied in increments.

Limitations of the Optimizing Method

It is of course impossible to quote a precise value for the drag coefficient limit which could be achieved by the use of this optimizing method. The value depends to a large extent on the original styling concept. A series of optimizations performed on medium-sized automobiles with non-complex styling has shown that the lower limit for these vehicles lies at approximately $c_D = 0.40$. If it is desired to make use of the potential illustrated in Fig. 6.1.32 and to proceed to a further reduction in drag coefficient, the optimizing method will have to be abandoned. Instead, specific low-drag configurations will have to be developed and will be bound to involve a different styling approach. Since the majority of current mass-produced automobiles can claim only a drag coefficient far above than the limit of $c_D = 0.40$ achieved by optimization, it would be appropriate at this time to focus on the potential of the optmizing method.

6.1.7 Mechanical Resistance

This term comprises all factors influencing fuel consumption which are due either to friction or to ventilation losses in the engine, which are caused by the engine having to transport considerable quantities of gases and liquids.

Friction losses may be due to the tires rolling on the road, to friction in the wheel bearings, the efficiency of final drive and transmission, and to friction in the engine itself.

Fig. 6.1.49 (2) illustrates the meaning of rolling resistance by comparing the rolling resistance over velocity curves of bias, bias-belted, and radial tires. We can see that especially in the velocity range which is legally permissible in the U.S. rolling resistance can be cut by up to 30 % by using the most suitable kind of tire. This trend is lost at higher velocities, but this is of no interest from the U.S. point of view.

One component factor of this rolling resistance is wheel friction. Its exact value is not given.

The resistance or, in other words, the efficiency of drivetrains differs according to whether automatic or manual transmissions are involved.

One compoment of the efficiency of automatic transmissions is the specific fuel economy loss which is typical for this kind of transmission (see Chapter 6.1.5). The efficiency of manual transmissions varies between 90 and 96 %, depending on whether the lubricant is hot or cold. Efficiency has a tendency to drop a little as engine speed increases; thus, for instance, it will drop by about 1 to 2 % over the engine speed range from 3,000 to 6,000 rpm.

Fuel economy is substantially influenced by internal engine resistance, i.e. the amount of energy needed to drive the engine itself. The more energy is used for this purpose the lower is the benefit which can be drawn from the energy potential of the fuel. Therefore it is necessary to keep internal resistance as low as possible.

Internal resistance is composed of power losses caused by friction, by ventilation, and by powering pumps. Friction losses are caused by bearings as well as by the movement of the pistons inside the cylinders. Ventilation losses are caused by the fact that considerable quantities of gas have to be transported through the engine which in some cases may be quite substantial. A lot of power is consumed by water, oil, and fuel pumps.

If the internal resistance of an engine increases, for instance, because of a drop in the oil temperature, it is necessary to open the throttle a bit further and feed more air/fuel mixture to the engine to ensure that the same amount of usable power is produced. This, of course, is impossible, along the full-load curve.

The extent to which friction influences the final power output of an engine depends on 3 factors, the first being engine speed, as Fig. 6.1.19 (see Chapter 6.1.4) shows, the second being the loading of the engine. The higher the load at a given engine speed, the lower the percentage lost by friction, for friction is independent of engine loading for a wide range. The third factor is the temperature of the engine. In a cold engine, the media to be transported are very sluggish, so the power consumed by friction is quite high. The influence of all other engine parameters on friction is limited to the extent of their influence on one of these three factors.

Fig. 6.1.19 related engine speed to engine friction resistance by giving the amount of power required to keep an engine turning at a given speed, which, in fact, constitutes an analysis of power losses through friction. We can see that, friction losses grow with the size of the engine.

In Fig. 6.1.19 we related the friction and ventilation loss at each given speed to the maximum engine performance at this speed. Expressed in round terms, engines have to produce first at each speed 30 to 50 % of their maximum torque output to overcome their own internal resistance, in other words, they need it only to keep rotating at that speed regardless of what performance they are producing.

The uninterrupted curves in Fig. 6.1.19 represent internal power losses over engine speed with the throttle open, whereas the dashed curves represent the same quantity with the throttle closed. We can see that there is not much difference, because with the throttle open large quantities of gas have to be transported, which causes ventilation loss, whereas with the throttle closed much work is required to generate a high vacuum between throttle and piston and to maintain that vacuum. In other words, the pressure-over-volume diagram in this case would show negative pumping work. Fig. 6.1.19 shows that in the low-to-middle engine speed range the intake cycle has a detrimental influence which is only overlaid at high speeds when the throttle is open and large amounts of gas are being transported, which then reduces more the performance of the engine than the vacuum generating.

From the closed throttle curves in Fig. 6.1.50 it can be seen that the 1.6 l engine has a relatively high friction loss. The big engine seems to be more affected by ventilation loss which is indicated by the fact that the specific loss of the 2.2 l engine is the same as of the 1.6 l engine in the open throttle case. This seems to indicate that this engine was originally designed to handle a higher throughput of gas than the present U.S. version. As a matter of fact, this same engine as supplied to other countries features a peak horsepower of 130 HP instead of 103 in the U.S.

Fig. 6.1.27 (2)(see Chapter 6.1.6) shows the way in which the amount of power consumed by operating a vehicle along the partial road load curve can be broken down into the amount of power required to overcome air drag, and the amount of power required to overcome mechanical resistance.

We can see that expressed in relative terms more power is required to overcome mechanical resistance as the velocity of the vehicle drops, whereas the amount of power required to overcome air drag increases with the velocity of the vehicle. In the example shown here, the point where the two lines intersect is located between 50 and 60 mph. In other words, the balance between aerodynamic and mechanical resistance is reached approximately at the top speed legally permitted on American highways. Consequently, if you want to improve Urban fuel economy you have to concentrate on mechanical resistance, whereas the Highway fuel economy is influenced to a considerable extent by air drag.

6.1.8 Power Consumed by Auxiliaries

Besides the equipment used for heating and cooling the passenger compartment, which will be dealt with in Chapter 6.1.9, there are a number of auxiliaries which are enumerated below. Power for these auxiliaries is generated directly or indirectly by the engine, which means that it is ultimately drawn from the fuel reserves of the vehicle:

Oil pump

Water pump

Fan

Alternator

Secondary air pump

Power steering pump.

In the following, we will investigate the power consumption of these units and the extent to which they influence fuel consumption.

Oil Pump

Oil pumps are installed to keep the lubricant in an engine under pressure and to supply oil to all bearings. Fig. 6.1.51 is a map showing the characteristics of the oil pump of the 1.6 l Rabbit engine. Power consumption is among the data given in this map. The fat lines indicate constant pump speed, the pump speed in each case being equivalent to the speed of the distributor shaft, i.e. to half the speed of the crankshaft. The interrupted curve indicates the way in which the oil pressure in the engine changes with the speed of the pump: After a quick initial rise, pressure remains constant because of the control valve which keeps oil pressure at a uniform level independently of the speed of the oil pump. The oil pressure is high enough to ensure that all bearings are sufficiently supplied with oil.

At the rated oil pressure and at an engine speed of 6,000 rpm, the maximum power consumption of the oil pump amounts to approximately 1 HP. It is clear from the graph that the quantity of oil moved by the pump becomes too great at point 1, where a kink in the delivery curve of the lubrication system marks the point where it becomes horizontal. Right on point 1, the quantity of oil supplied is still just right, but the farther you follow the horizontal line away from that point, the more extensive is the discrepancy between the quantity of oil supplied and the

quantity actually required. Assuming as a first approximation that the oil pressure required over the entire engine speed range is 6.2 bar, and postulating that the quantity of oil required hardly varies at all with the engine speed, even if the fact that the oil temperature at high and low engine speeds differs is taken into account, it is conceivable that the power consumption of the oil pump may be cut by 0.7 HP at engines highest speed simply by rendering the speed of the oil pump independent of the engine speed and powering it, for instance, by an electric motor.

In actual fact, however, it is not possible to save quite as much power. First of all, there is the efficiency of generator ($\sim$ 0.5) and pump motor ($\sim$ 0.8) to be taken into account; secondly, it is quite a lot to assume an average pump speed of 1,000 rpm, which corresponds to an average engine speed of 2,000 rpm, considering, for instance, the average speeds attained in urban driving, see paragraph "Alternator" in this chapter. In other words, powering the oil pump by an electric motor would be profitable only in highway driving, and only very slightly at that. And it is quite probable that in Urban driving and electric oil pump would prove to be inferior because of the efficiency of generator and motor. Especially after a through analysis of cost and benefit, taking into account the additional expense involved, it seems safe to recommend to leave the oil pump as it is generally designed today, being powered directly by the engine.

Water Pump

The water pump circulates the coolant through the engine. If there is a high energy turnover, the coolant throughput must be large as well; if there is a low energy turnover, the throughput is small. It is logical, therefore, to have a direct link between water pump and engine speed. In larger engines, the power consumption of the water pump may be related to the engine speed in the manner described in Fig. 6.1.52 (2).

So there is not much energy to be saved by modifying the water pump. One point to be considered, however, would be that with the water pump directly coupled to the engine the coolant throughput at every engine speed should be such that in all parts of the engine it will suffice to absorb the excess heat generated by the highest power output which is possible at that speed. This means that it is possible to cut the energy consumption of the water pump by a small amount by installing a governor which will reduce the speed of the pump if the engine, for instance, is operating a partial road load.

Fan

Another point illustrated by Fig. 6.1.52 is that a considerable amount of energy can be saved by disconnecting the fan from the engine. In addition to the various alternatives enumerated in this figure, it is possible to have the fan powered by an electric motor of its own. This design has been used with great success in the water-cooled engines made by Volkswagen. The amount of energy which can be saved in this way is larger even than what is indicated in Fig. 6.1.52 Estimated according to the method described in the next paragraph, the power consumed by the fan of a Rabbit not fitted with air conditioning is a mere 0.1 HP on an average.

Alternator

To make sure that the necessary amount of electric energy is supplied at all times it is necessary to have an alternator driven by the engine. The amount of power generated depends on the size of the alternator, the transmission ratio, and the engine speed range. Alternators should always be dimensioned to ensure that the amount of power required by the vehicle at any given moment will be available. The more electric energy is consumed by a vehicle, the larger the alternator, and the larger the amount of power consumed by it.

Alternator size and drive transmission ratio are determined by computing the power balance of a vehicle, i.e. by comparing the possible power consumption of all electric consumer units installed in the vehicle to the maximum amount of power which the alternator will supply in UDC operation. Urban driving is used as a basis for these calculations because the output of the alternator is lowest then.

What is called the 'theorectical power consumption' is computed by adding up the maximum power consumption of each consumer unit multiplied by its average period of activity. The latter factor is based on previous experience and amounts to 100 % in most cases. Table 6.1.11 is a typical list of all electric units in a Rabbit. Positions 8 and 9 as well as the fact that position 7 is higher than position 6 indicate the existance of consumer units which will be dealt with separately in Chapter 6.1.9.

We can see that the 'theoretical power consumption' of all standard auxiliaries amounts to 34.4 A. Including air conditioning brings that figure up to 61.8 A. To arrive at the final capacity of the alternator, we have to compute its urban generating performance.

The urban generating performance of an alternator signifies the average power output while driving through a city. The data on which this figure is based are generated during test runs in the traffic of large towns,- in the course of which the engine speed is recorded by the second. Using the delivery curve of the alternator concerned, and taking into account the transmission ratio of its drive, the power supplied by the alternator at all speeds recorded can be computed.

Classifying all data into 75 different groups and computing the relative frequency of each group provides the means by which to calculate the average power output of alternators.

Fig. 6.1.53 shows engine speed curves obtained from very similar engines installed in two different vehicles, a light duty truck (interrupted line) and a passenger car (uninterrupted line).

Due to high power requirements, it is easy to see that the general engine speed level of the light duty truck is higher than that of the passenger car.

The 'theoretical power consumption' of all consumer units is then subtracted from the urban generating performance. If the result of this subtraction is -15 A or more, VW consider the capacity of the generator to be sufficient. If the result is less than -15 A, a larger generator has to be used. There is no objection to negative figures in this case because the urban generating performance is measured under operating conditions which allow only the lowest generating output. As it is reasonable to assume that other driving modes or power requirements will occur in between times, occasional power deficits may be expected to be covered by the battery.

The amount of energy consumed by the alternator can be computed from the assumption that on an average the maximum power consumption is by no means equal to the real figure, so that the urban generating performance of the alternator will generally suffice to supply the amount of electric power demanded by the vehicle.

In other words: The extent to which the alternator influences fuel consumption is mainly dependent on its urban generating performance, divided by the efficiency of the alternator, which in today's units ranges around 0.5 under urban driving conditions (see Fig. 6.1.54).

To arrive at the extent to which individual consumers contribute towards the alternator's influence on fuel economy we have to keep in mind that the urban generating performance of the alternator amounts to approximately 70 % of the 'maximum consumption' figure, i.e. that the average consumer unit participates in the general consumption of power at a rate of about 70 % of its maximum consumption. But as the efficiency of the alter-

ternator demands mechanical work amounting to 100 % to produce electric work amounting to 50 %, the average electric consumer participates in the general power consumption at 140 % of its maximum power consumption.

Secondary Air Pump

Fig. 6.1.52 shows the power consumption of the secondary air pump over engine speed. Although this pump did play a certain role in the investigations performed under this Contract we think that its significance will decline considerably in the long run. Therefore, it seems sufficient to assume a relationship between the power consumption and the speed of the secondary air pump such as the one given in Fig. 6.1.52, although this relationship is surely fairly close to the upper limit of anything which might occur in practice.

Power Steering Pump

Compared to the systems which are considered before, the extent to which steering aids (or power steering) influence fuel economy can be diminished to a considerable degree. Zahnradfabrik Friedrichshafen AG have just presented a comparison of various power steering systems (Table 6.1.12) which indicates that the increase in fuel consumption caused by introducing power steering in an AUDI 5000 (inertia weight 3,000 lbs) can be reduced from 0.17 gal/100 miles to 0.12 gal/100 miles. They are confident that by continuously developing this system it is possible to reduce this figure even further to 0.04 gal/100 miles.

When considering the progress which has been made in this respect one should keep in mind that AUDI already have a steering aid whose power consumption is only about 61 % of those commercially used in the U.S. today.

This power saving is due to the fact that in this system the traditional vane-type pump is replaced by a radial piston pump whose hydromechanical efficiency is better. Moreover, the permanent throttling loss which occurs at the open-centred control valve is eliminated by installing a closed-center control valve instead. Here, throttling losses occur only if and when the power steering is actually operating. Moreover, a pressure storage unit is installed to strike a better balance between the power requirements of the power steering system and the varying power supply of the engine.

We expect, however, that it will only be possible to cut the fuel consumption increase down to 0.04 gal/100 miles once all leakage has been eliminated from the control valve; but this is still a thing of the future.

6.1.9 Energy for Heating and Cooling

There are two kinds of heating system, one being the standard system which uses waste heat from the engine which is present in the exhaust gas and in the coolant for heating the interior of the car via a heat exchanger. As this system uses heat which is available free of charge it is installed in practically all vehicles now in existence. Energy here is needed to power the fan which doubles as a ventilator conveying fresh air into the interior of the car, and which usually consumes between 50 and 100 Watts. This power is drained from the supply system of the car, and it is therefore a factor in designing the capacity of the alternator.

The second kind of heating system is nearly in all cases used as an auxiliary, being activated only if outside temperatures are excessively low and the warmth gained from the waste heat of the engine is insufficient. These heating units burn fuel, that is why they cause a direct increase in fuel consumption. Running without interruption, these heaters will use between 0.25 and 0.65 liters per hour in a passenger car, depending on the vehicle size and ambient temperature. Of course, these auxiliary heaters also use electric energy to power fans, fuel pumps, and ignition systems, consuming between 150 and 250 Watts, the latter being the power consumption of very big passenger car heaters.

Estimating the extent to which these units influence fuel consumption can be done along the lines laid down in Chapter 6.1.8: When determing the extent to which auxiliaries influence fuel consumption, one has to keep in mind that the total amount of power which all auxiliaries divert from the power output of the engine is 40 % higher than their maximum rated power input (see Chapter 6.1.8 paragraph alternator). Although this procedure tends to present individual auxiliaries in an unfavorable light occasionally, the average on the whole is correct.

The possible increase in fuel consumption caused by air conditioning in motor vehicles is much higher. First and foremost, there is the compressor itself, which is activated or deactivated according to whether cooling is required or not. In operation, the compressor speed is directly proportional to the engine speed. By rule of thumb the power required to operate the air conditioner is directly proportional to the engine speed, growing by 0.57 kW with every additional 1,000 rpm.

Moreover, there are other consumers attached to an air conditioning plant, such as the electric clutch which connects and disconnects compressor and engine. Then, more powerful ventilators are required, and the cooling fan has to be redesigned because the condenser of the air conditioning unit is installed in front of the radiator and diminishes the flow of cooling air. Increasing the dimensions of the fan ensures that sufficient air reaches both the radiator and the condenser at all times.

If all these factors are taken into account in the way employed in connection with the heating system, the amount of electrical power diverted by an air conditioning unit is about 500 W (see Table 6.1.12).

In Fig. 6.1.55, we have used EPA certification test data of 1977 to draw up a diagram representing the extent to which the fuel consumption of cars of 3,000 lbs inertia weight fitted with automatic transmission is influenced by an air conditioning unit. According to this diagram, the Combined fuel economy of all these vehicles deteriorates by 1 to 1.5 mpg.

6.1.10 Fuel Requirements

The hydrogen/carbon ratio of a fuel and the structure of its molecules will influence its calorific value, its speed of reaction, its knock rating, and the range of air/fuel ratios in which it may be used. Fuel economy is favorably affected by high reaction speeds, the possibility of operating on lean mixtures, and high compression ratios.

In pure hydrocarbons there is a linear relationship between the hydrogen/carbon and the air/fuel ratio. This means that in a stoichiometric mixture, changing the hydrogen/carbon ratio by 1 % will cause the air/fuel ratio to change by 1 % as well. The hydrogen/carbon ratio of commercial fuels, however, which consist of a mixture of various hydrocarbons, commonly varies only a little from its general level, which is around 13: 87. The difference in the H/C ratios of the various hydrocarbons which may be used as fuels amounts to approximately 20 % (e.g. methane - isooctane). If other fuels are used, the air/fuel ratio is influenced more noticeably. Thus, for instance, the amount of air required to produce a stoichiometric mixture is 6.4 kg per kg of methanol and 34 kg of air per kg of hydrogen, whereas the amount of air required for a kg of commercial gasoline is about 14.5 kg.

Uneven mixture distribution may be caused by incorrect fuel metering as well as by incomplete evaporation of the fuel as it enters the intake manifold. Fuel droplets or a fuel film flowing along the wall of the

intake manifold cannot be distributed evenly. For this reason, the boiling point of the fuel constitutes the dominant factor influencing the uniformity of mixture distribution and, therefore, fuel economy, emissions, startability, and acceleration performance. If the evaporation behaviour of fuels could be optimized, all these problems would easier to be solved.

Fuel economy is also decisively influenced by the knock characteristics of the fuel used. Knock is a phenomenon caused by irregular combustion of the ignited fuel leading to a loss of power and a deterioration of fuel economy. In extreme cases the engine may even be destroyed entirely.

Knock is caused by spontaneous combustion of the mixture in the cylinder which has not yet been reached by the flame front, i.e. by the fact that the as-yet unburned part of the mixture is subjected to too much strain by the burning part of the mixture (temperature and pressure) and by the preliminary reactions taking place in that area.

Knock is mainly affected by the octane number of the fuel used. High octane numbers mean small danger of knocking.

Knock is directly influenced by the H/C ratio of the fuel and especially by the structure of the hydrocarbon molecules. Larger fuel molecules of the same molecular structure will cause the knock properties of the fuel to deteriorate. From low to high, the octane numbers and, therefore, the knock resistance of some hydrocarbons increases in the order named: n-paraffins, olefins, cyclo-paraffins, iso-paraffins, aromatic hydrocarbons.

Engine map designation areas of optimum fuel economy or optimum emissions can be narrowed down by knock in some fields. Ranges combining high loads and low engine speeds are especially critical, as are the ranges combining high engine speeds and high loads, the former occuring during acceleration, the latter when driving at high speeds or up steep inclines (see Fig. 6.1.56). Under normal operating conditions, low engine speeds under high loads are likely to endure for relatively brief periods of time only, which means that this condition is not likely to endanger engine durability. In this case, the only adverse aspect of knocking is that its noise affects comfort adversely. Prolonged knock at high engine speeds adversely affects fuel economy and, from a certain intensity the durability of the engine as well. Knock may be prevented by reducing the compression ratio, setting back the spark advance, or by using fuels of higher knock-resistance. Only the last-named way will not increase fuel consumption.

To some degree, knock also depends on atmospheric conditions. Any change in atmospheric pressure produces a change in air density, which in turn affects the mean effective pressure. The tendency to knock decreases with atmospheric pressure, whether the latter is reduced by weather conditions (5 %) or by altitude (up to 30 %); provided that in the latter case, the air/fuel ratio is normally kept constant by means of an altitude compensator.

Air temperature directly influences the temperature of the cylinder charge. The mean effective pressure is influenced indirectly via air density. Knock increases with the air temperature: An increase in the cylinder charge inlet temperature (air temperature) will produce increased knock despite the fact that the mean effective pressure decreases as the air temperature increases (see Fig. 6.1.57 (4)).

An increase in humidity means that more water vapor (a triatomic inert gas) will enter the combustion chamber, thus reducing the likelihood of knock.

The parameters dealt with in the following paragraphs influence knock as well. These parameters, however, if used to lower the incidence of knock, will mean a loss of fuel economy.

The air/fuel ratio is one of the parameters influencing knock. However, the maximum knock incidence occurs at an air/fuel ratio of 10 to 15 % rich (see Fig. 6.1.58 (4)) wich is nearly the maximum of mean effective pressure too with a given air amount. Generally, the tendency to good fuel economy and low emissions cause that air/fuel ratios as rich as this will occur only close to full load. As the tendency to knock drops with decreasing load and decreasing temperatures the low air/fuel ratios occuring for instance, at part load and idle of the cold engine, do not influence knock. There is, however, a great danger of knock everywhere around the full-load curve as the air/fuel ratio in this range is adjusted to maximum power output.

Uneven mixture distribution involves an uneven distribution of fuel quality as well. For instance: Low and high-boiling fuel components may have different octane numbers. If a cylinder is charged with fuel containing a high percentage of low-octane components it will be more prone to knock than another cylinder receiving a different charge. For this reason, it is necessary to ensure that fuel additives used to improve the knock behavior of individual fuel components are distributed throughout the intake manifold as evenly as these components themselves.

If this should prove impossible, and if, therefore, the individual cylinders are charged with mixtures of varying knock resistance, it

may happen that the spark advance of all cylinders has to be adjusted to suit the most critical cylinder, which in turn makes it impossible to optimize fuel economy.

If the compression ratio of an engine is increased in order to improve fuel economy this will result in increased temperatures and pressure occurring during polytropic compression as well as during combustion. That part of cylinder charge which is in danger of causing knock will be subjected to higher temperatures and pressure during the compression stroke, so that reactions initating knock may take place. It may then become necessary to use more knock-resistant fuel.

More advanced ignition is more likely to produce knock (see Fig. 6.1.59 (4)). As ignition is delayed, the final combustion pressure will drop and with it the temperature reached by the part of the cylinder charge which is burned last. Thus, the likelihood of knock is reduced. In most cases, however, this solution entails a loss of power. Depending on the type of engine concerned, it becomes impossible to extract the maximum amount of work from the engine either at low or at high engine speeds. This again means a loss of fuel economy.

Usually, spark advance is the same for all cylinders. If the mixture is distributed unevenly, however, spark advance of at least one cylinder will have to be set back in order to avoid knock. This means that power output and fuel economy will be adversely affected not only by bad mixture distribution but also by less-than-optimum ignition timing.

As we have seen, the quality of the fuel used influences fuel economy very drastically. But as automobile manufacturers are unable to influence fuel quality, they have to live with whatever fuels are commercial.

6.1.11. Engine Design Parameters

Number of Cylinders

Although there is no doubt that a higher number of cylinders entails a higher fuel consumption there are no test results corroborating this assumption, therefore it is impossible to quantify this influence. It is a well-known fact that increasing the number of cylinders leads to a decrease in thermodynamic losses because of the change in the surface-to-volume ratio. It is just as well known, however, that a higher num-

ber of cylinders entails higher losses through friction, and that the thermodynamic efficiency gain is more than compensated by these losses. So, a higher number of cylinders cannot but lead to higher fuel consumption. This relationship will be all the more pronounced as the specific power output of the engine decreases, or, in other words, after increasing the number of cylinders the resultant deterioration of the fuel consumption should be worse in UDC than in HDC because the percentage of the total power output which is consumed by friction is higher if the specific power output is low. However, we cannot prove this by our test results, either, because in the phenomena associated with the transition from a 4-cylinder to a 5-cylinder engine it is impossible to distinguish between the influence of the number of cylinders and that of displacement. We do not even know enough to give an educated guess, because the fuel economy differences between the 1.3 and the 1.6 l engine cannot be used as a standard for determining the influence of displacement because the design of the 1.6 l engine entails a friction increase which is independent of the increase in displacement.

This being so, there is at present no alternative but to accept a purely qualitative statement to the effect that a higher number of cylinders leads to fuel economy losses.

Displacement

In Fig. 6.1.60 we have related the displacements of the engines mentioned in Chapter 6.1.4 to the fuel economy ratings given in Fig. 6.1.17 in the same Chapter. There is no doubt that the scatter bandwidth of fuel economy ratings obtained from the 1.6 l engine is large, because this engine, one of the most important engines manufactured by VW, is installed in a wide variety of vehicles. Yet there is a trend indicating that fuel consumption drops with engine displacement. Although this trend is not as marked as this graph indicates - after all, low-displacement engines are installed in vehicles of smaller mass - there can be no doubt that it is there (see Chapter 6.1.4).

Modifying the displacement of an engine exposes fuel economy to a number of opposing influences. As it is necessary to use the same vehicle for testing engines of different displacements, the same power output is required, for instance, to maintain a certain velocity at part throttle. A high-displacement engine will maintain this velocity at a lower speed as a low-displacement engine, and there is a tendency for higher engine speeds to be associated with higher friction losses.

All this is counteracted by the following factors:

1) As the high-CID engine has larger geometric dimensions its friction losses are higher as well. As we shall show a little bit farther along, this tendency is more marked in engines enlarged by increasing the bore than in engines enlarged by increasing the stroke.

2) Compared to the low-CID engine, the high displacement engine will be throttled down more to produce the same power output. This is indicated by the two dot-dash lines in Fig. 6.1.18 (see Chapter 6.1.4) which represent the throttle angles of the high and of the low-CID engine producing the same power output. The final compression pressure of the bigger engine is lower than that of the smaller engine producing the same power output, which means that the bigger engine is not so close to its point of optimum efficiency which, as will be shown later on in this Chapter, necessarily leads to higher fuel consumption.

It can be said that as a rule increasing the displacement of an engine will entail a loss of efficiency, because the negative factors will outweigh the other.

Tests which could be used to judge the influence of displacement alone are not available. All data available always refer to the influence of two or more parameters together, so that it is impossible to make any quantitative statements conerning the aspect dealt with in this Section.

Stroke/Bore Ratio

The stroke/bore ratio was in the focus of intense general interest in the early Sixties. The point was to find out whether it was sensible to follow the trend towards shorter and shorter strokes, or whether there could be any ulterior points of view, such as that of fuel economy, from which it might seem advisable to abandon this trend. For this reason, extremely divergent stroke/bore ratios ranging from 0.92 to 0.4 were tested, all of which were under-square. The two most significant findings produced by these tests are shown in Figs. 6.1.61 (5) and 6.1.62 (5). The first figure illustrates the relationship between friction, stroke/bore ratio, and engine speed. Here, friction is represented as that percentage of the mean effective pressure recorded which is used to overcome friction. The reason why this figure indicates that the long stroke engine loses more energy through friction is given in Fig. 6.1.63 (2): In the long stroke engine, the velocity of the pistons is higher, which means that there is more friction as well.

The curve in Fig. 6.1.62 indicates specific fuel consumption over engine speed at various stroke/bore ratios. We can see that at low and middle engine speeds the fuel consumption of long stroke engines is lower than that of short stroke engines in spite of their high friction losses. This is obviously due to the large area of the surfaces which radiate heat, which in engines with a wide bore are more extensive in the times when high temperatures occur, i.e. in the combustion chambers.

At low and middle speeds where friction is not yet dominant, large surfaces of this nature entail extensive heat losses which influence specific fuel consumption. With increasing engine speed the differences in friction losses become so pronounced that, after overcoming the influence of the heat-radiating surfaces, they finally cause the specific fuel consumption of long stroke engines to increase beyond that of short stroke engines.

Fuel economy also depends to some extent on the influence of bore and stroke on the dimensions of the engine and on its mass. The significance of this factor lies in the fact that the inertia weight of the vehicle depends on it both directly and indirectly: The weight of the engine has a direct influence, whereas the influence of its dimensions is indirect because the size of the engine compartment and with it the mass of that particular section of the vehicle depend on them. Figs. 6.1.64 (5) and 6.1.65 (5) indicate the relationship between stroke/bore ratio and engine dimensions. The data used in these figures are derived from design analyses. They indicate that oversquare engines are superior in this respect as well.

Compression Ratio

Compression ratio is among the engine design parameters whose influence on fuel consumption has been recognized for a very long time. At the time of writing, we are not aware of any studies dealing with the influence of compression ratio on fuel economy under non-steady-state, i.e. driving cycle conditions. Steady-state tests have shown that the compression ratios of between $\varepsilon = 7$ and $\varepsilon = 11$ which are common in today's engines have an approximately linear influence on fuel consumption when compared at optimum efficiency, i.e. at $p_e/p_{emax} = 0.9$. This means that fuel consumption will drop by about 5 % as the compression ratio increases by 2 units (see Fig. 6.1.66). If you reduce the power output at which the various compression ratios are compared, you will find (see also (6)) that the significance of the compression ratio is augmented: The higher the compression ratio, the lower the percentage by which specific fuel consumption increases as the load is reduced (see Fig. 6.1.67). As the loading used in fuel economy driving cycles is generally below 90 % it is all the more urgent to apply this finding and increase the compression ratio. It is, however, a sad fact that it is impossible to increase the compression ratio at will because of the knock which occurs under full load.

This particular condition might be relieved by introducing compression ratios which vary with the engine loading, which would result in a specific fuel consumption over throttle opening angle curve like curve a) (7) in Fig. 6.1.67. The possibility of eliminating knock would justify compression ratio changes as drastic as these; because the relationship between throttle opening angle and knock intensity is as close as it is represented in Fig. 6.1.68 (7).

Fig. 6.1.69 (7) shows the fuel consumption improvement obtainable at constant speeds by variable compression ratios.

Combustion Chamber Configuration

In the past, VW have run a large number of tests to optimize combustion chamber configurations so as to obtain optimum emissions, fuel consumption, and knock. Moreover, quite a number of outside publications have dealt with this aspect in recent years, so that we are very well aware of what can be done by optimizing combustion chambers.

It has been known for a long time that there is no connection between combustion chamber configuration and CO emissions, the latter being nearly exclusively dependent on the air/fuel ratio. It is also known traditionally that compact combustion chambers with low surface-to-volume ratios are outstandingly suitable for reducing hydrocarbon emissions and knock and for attaining high compression ratios and good fuel consumption. Small surfaces mean small quench zones, which in turn entails low hydrocarbon emissions in steady-state operation. Simultaneously, compact combustion chambers are associated with a short flame travel and a brief combustion time, so that preliminary reactions, if any, cannot develop into knock.

Fig. 6.1.70 (7) shows the relationship between combustion chamber configuration and combustion time. Compared to the two cylinder head chambers, the combustion time in a cup-type chamber in the piston is somewhat shorter because, due to the comparatively high temperature of the combustion chamber walls which are not in contact with any coolant combustion in the piston is swifter and more thorough. However, the time of combustion is by far shortest in what is called the AS chamber, which is due to its nearspherical shape as well as to the wide quench zone in the left part of the piston which creates intensive turbulence in the combustion chamber. Here, the combustion time corresponds to a mere 28° crank angle, whereas in the wedge-type combustion chamber it corresponds to all of 41°.

To sum up, we may say that the combustion chamber configuration is influential in three respects, compactness, the turbulence of the mixture, and the temperature of the combustion chamber walls.

Under these aspects VW initiated studies aimed at optimizing combustion chamber configurations. Fig. 6.1.71 shows the major configurations on which these studies were based, whereas Table 6.1.14 lists all major geometric data of these chambers. The curves in Figs. 6.1.72 and 6.1.73 represent mean effective pressure versus engine speed and specific fuel consumption at full load versus engine speed, respectively. The differences indicated here go up as high as 5 to 6 % as far as both mep and fuel consumption are concerned. As the develop-

ment trends of the mep curves and the specific fuel consumption curves are approximately similar it seems permissible to conclude that the variations recorded are due to differences in the combustion process itself and not to differences in the charges.

All these experiments are, of course, valid only if the air/fuel ratio is kept constant at all times and the ignition advance is kept at optimum. As we can see from these figures, the cup-shaped piston with swirl has some outstanding properties.

At present, we have no way of explaining why the non-squish cup-shaped piston without swirl is hardly better or even worse than the basic chamber, whereas the one with swirl is far and away the best. This is aggravated by the fact that the cup-shaped piston alone features the highest quench-area-to-piston-area ratio (see Table 6.1.13) so that its turbulence should be quite high even without any squish effect, making for high combustion speeds and good fuel economy. And, finally, combustion in this kind of piston really ought to be fast, considering that the walls of the combustion chamber are hot. Fig. 6.1.70. indicates the same trend.

Camshaft

Up to now, the automotive industry used to design camshafts mainly along lines of consideration aiming at obtaining maximum torque at any given speed and at satisfying certain mechanical criteria as well, the latter being especially the durability of the valve gear and its ability to work well at very high speeds.

The full-throttle curve over speed is affected by a variety of factors which depend on the characteristics of the engine. If the peak horsepower of the engine is to be especially high the camshaft must be designed to ensure that the cylinders receive maximum charges at high engine speeds; if the engine is to be outstandingly elastic, one must aim for maximum charges at low engine speeds. One can either opt for one of these two alternatives and use the appropriate type of camshaft, or one can compromise and use in-between solutions.

In any case, it is advantageous to have cams with steeply inclined contours to ensure that the process of opening the valve is completed as swiftly as possible once it has started; the same applies to the process of closing the valve. The only limitations to increasing the inclination of the cam contours are purely mechanical in nature.

Once the inclination of the contours has been decided upon, the full-load characteristics of the camshaft are settled by selecting the proper valve timing.

Repeated studies have established the fact that the automotive industry is basically right in adapting the design of camshafts to the desired full-load curve of the engine, because solutions found in this way will always satisfy fuel economy considerations as well.

Figs. 6.1.74 and 6.1.75 show the results of one of these studies dealing with a total of three different camshafts, one featuring negative valve overlap, one a slight positive overlap, and one a somewhat more extensive overlap. These three camshafts were installed in different positions relative to the crankshaft, which resulted in the various valve timings shown in the two figures (see also Table 6.1.14).

The top of each graph shows the way in which fuel economy was found to change with the valve timing. The actual test results are fuel economy ratings determined by varying the air/fuel ratio until the camshafts in their respective positions had reached the attitude of optimum fuel consumption. Fuel economy figures were determined at part throttle, i.e. at 31, 62, and 75 mph in Fig. 6.1.74, and at 31 mph in Fig. 6.1.75.

The immediate result of this test is that all fuel consumption changes caused by modifying camshafts and valve timings are no more extensive than the dispersions which are normal in measurements of this kind. This means that it is diffcult to make any statements at all, and if a statement is made it will stand on rather wobbly foundations.

Still, it is possible to draw three conclusions from all this, the first one being that introducing a large valve overlap and opening the outlet valve early will bring about a deterioration in fuel economy, whereas advancing the inlet valve closing time will improve fuel economy. Camshaft positions I, II and III on the graph show that it is no good to have much valve overlap and to advance the outlet valve opening time. In these positions, the inlet closing time remains constant, whereas valve overlap increases, resulting in an unmistakable loss of fuel economy. From position IV to position VI, valve overlap is reduced while the outlet opening time remains constant and the inlet valve is closed at an increasingly later time; there is no trend here which is clearly distinguishable. It is possible to recognize a very slight improvement in fuel economy as the inlet closing time advances from position VII to position IX in spite of the outlet closing time being advanced as well; there is a further slight improvement from position X to XI for the same reason; and there is an unmistakable improvement from position XII to position XIV.

That much valve overlap may lead to extensive fuel economy losses is susceptible to explanation in so far as any increase in valve overlap is generally accompanied by an increase in hydrocarbon emissions, which indicates that some fuel is lost by being transferred unburned from the inlet to the outlet port. Advancing the outlet opening time too far causes loss of pressure in the cylinders and some loss of energy as well.

Advancing the inlet closing time may contribute towards improving the fuel economy because it causes a drop in the intake vacuum by wider open throttle if the power output of the engine is not affected. So the negative work in the engine during the charge cycle is reduced.

There are a number of publications past and present which deal with the problem of extended expansion (for instance (8)). We do not know if this idea has hitherto found its way into concrete application in a mass-produced vehicle, but at least theoretically it may well lead to considerable fuel saving.

In the following, we will explain briefly the theory on which the idea of extended expansion is based. Fig. 6.1.76 compares by means of a pressure/volume and a temperature/entropy diagram the thermodynamic process in an ideal spark ignition engine (constant-volume cycle) and in an extended expansion engine. Figures related to the extended-expansion cycle are identified by an asterisk*.

Final compression pressure and final combustion pressure are not affected by the transition from the constant-volume to the extended-expansion cycle; in other words, the same fuel with the same octane number may be used in both cases. Neither does the compression ratio change; it remains

$$\varepsilon = \frac{Vc + Vh}{Vc} = \frac{Vc^* + Vh^*}{Vc^*}$$

There is, however, a difference between the two cycles in that in the normal constant-volume cycle the expansion ratio δ is the same as the compression ratio ε , thus:

$$\delta = \varepsilon = \frac{Vc + Vh}{Vc}$$

whereas in the extended-expansion cycle the expansion ratio is

$$\delta = \frac{Vc^* + Vh}{Vc^*}$$

as a result of which additional useful heat is produced to an extent which is directly proportional to the increase in the area of usable work in the temperature/entropy diagram, which is bordered by the lines connecting 4 - 4* - 5* - 1*.

If we now assume the efficiency of the ideal cycle (ideal gas; no change in composition; constant polytropic exponent $\varkappa$; no disassociation) to be η_v; if we further assume the $\delta/\varepsilon = \varphi$ ratio to express extended expansion as a factor, and that $P_3/P_2 = \psi$ expresses the increase in pressure relative to the calorific energy introduced in the process, the efficiency of the normal constant-volume cycle is

$$\eta_v = 1 - \frac{1}{\varepsilon^{\varkappa-1}}$$

and that of the extended-expansion cycle is

$$\eta_v = 1 - \frac{1}{\varepsilon^{\varkappa-1}} \cdot \frac{\left(\frac{\psi}{\varphi^{\varkappa-1}}\right) - \varphi + \varkappa(\varphi - 1)}{\psi - 1}$$

It is obvious that the value of the additional term in the formula expressing the efficiency of the extended-expansion cycle is always below 1, which means that this term indicates that the efficiency of the extended-expansion cycle must be higher.

Only if $\varphi = 1$, which is the case if there is no extended expansion, the additional term will be equal to 1, so that the two efficiencies become the same.

These findings have been put to the test in several engines, which resulted in remarkable fuel savings. So it would indeed pay to reconsider this idea, keeping in mind, however, that this would reduce the peak horsepower of the engine in question.

Applying this process in practice would involve advancing the inlet valve closing time to a relatively large extent, keeping in mind that actual hardware can never fully match a theoretical process, and increasing the expansion ratio by reducing the space of the compression chamber. As the process of expansion is changed it is possible to gain a bit more by retarding the outlet valve opening time. However, in this process as well as in power output trade-offs one should never lose sight of the fact that the sum total of expansion and exhaust losses must be kept at a minimum. Consequently, this is another case where valve timing must depend on engine speed.

6.1.12. Engine-CID-to-Peak-Horsepower Ratio

Fig. 6.1.77 shows the mean values of all fuel economy tests run under this Contract, including those of Fig. 6.1.17, entered over their

respective engine-CID-to-peak-horsepower ratios, demonstrating that there is obviously no connection between fuel economy and engine-CID-to-peak-horsepower ratio. The highest and the lowest fuel economy figures come rather close to having the same CID-to-peak-horsepower ratio, and the fuel economy figure associated with the lowest CID-to-peak-horsepower ratio is higher than one of the four highest ratios.

The number of different symbols in this graph, the meaning of which is explained in Fig. 6.1.77, indicates that there must be other influences which cause fuel economy to vary. Inertia weight, for instance, is one factor which is surely significant. Apart from the cross symbol, which identifies a 5-cylinder engine, and from the CID/HP figure of 0.92 which belongs to a typical sports car, all comparable points are located within the same scatter band. It is especially the results obtained from inertia weight of 2,250 and 3,000 lbs which show that even within one and the same inertia weight class there is no correlation between CID/HP and fuel economy.

6.1.13 Engine Adjustments

In engine operation, there are five parameters theoretically capable of exerting an extensive influence on fuel consumption, namely three major parameters, throttle opening, air/fuel ratio, and spark advance, and two minor parameters, EGR ratio and intake air temperature.

Throttle Opening

Fig. 6.1.78 is an engine map, the engine in question being Modification Code 16. It is not very complicated to recognize from this graph the way in which specific fuel consumption is influenced by the throttle angle: Best values in WOT operation. Fig. 6.1.79 is another engine map illustrating the same phenomenon by showing throttle angle over absolute fuel consumption. That specific fuel consumption is at its best at wide throttle angles is because the thermodynamic efficiency of an engine increases with the initial pressure prevailing in the cylinders at the beginning of the compression stroke. If the throttle is closed and the engine is throttled down, the initial pressure drops and the thermodynamic efficiency with it.

As a rule, therefore, efficiency improves as the throttle is being opened. However, there is one exception to this rule: The WOT range close to the full-load line, where any further opening of the throttle will lead to a deterioration of

efficiency. This does in no way affect the basic fact that efficiency and throttle angle are related; the point is merely that in the full load case a relatively rich air/fuel ratio is used to obtain maximum performance and maximum torque.

Contrary to what is said on page 132 of this Chapter, the point of maximum torque at full throttle on an air/fuel ratio graph is by no means close to the point of minimum fuel consumption, for in this case power output is limited not by maximum absolute fuel consumption but by the maximum possible air throughput, which again is highest at air/fuel ratios < 1 because more fuel evaporates in this case, making for an improved cooling effect of the intake air.

The lines of constant fuel throughput per hour and of constant throttle angle are nearly parallel over prolonged distances at part load. This is because air throughput is determined by the throttle angle, save for the ranges of high load and low engine speed, and because every effort is made to keep the air/fuel ratio constant throughout the engine map and close to the line of minimum fuel consumption, the only exception being the full-load range.

Air/Fuel Ratio

In the early Seventies, a number of papers, (9) among them, mentioned quite rightly that the optimum efficiency of spark ignition engines is found at air/fuel ratios quite different from those which were assumed to be ideal until then. Many of these statements went unheeded because the problem of exhaust emissions dominated at that time. Now, with fuel economy prominent in our minds, these old ideas should receive some general attention again.

The principles governing the relationship between fuel economy, or efficiency, and air/fuel ratio are shown in Fig. 6.1.80 (2). However, the problem is more complex than it appears from this figure, for the objective is to produce a certain power output at the lowest possible fuel throughput $\dot{B}$ (g/h). Fig. 6.1.81 (9) is a graph showing fuel throughput over air throughput at different mean effective pressures but at constant engine speed. This graph shows that the lowest absolute fuel consumption occurs where the mean effective pressure curve touches its horizontal tangent, and not where the vertical tangent comes into contact with the lines of constant specific fuel consumption. Yet it is just this point which has hitherto been the objective of most fuel economy optimizations, for what is being done is to keep the throttle angle, i.e. for instance, the air throughput $\dot{L}$ (kg/h), constant while varying the air/fuel ratio until fuel consumption has reached a minimum.

That this is not the way to bring fuel consumption to its true minimum is palpably demonstrated by the fact that minimum specific fuel consumptions found in this way do not coincide with the points cf maximum mean effective pressure, which is, as Fig. 6.1.88 9 shows, located more towards the rich mixture side.

This, however, cannot be the case because we have

$$b_e = \frac{\dot{B}}{Ne}$$

with b_e representing specific fuel consumption expressed in g/HPh, $\dot{B}$ representing fuel throughput, and N_e representing the power output expressed in HP. Moreover, we have

$$Ne \sim Pe \, . \, n$$

with p_e representing mean effective pressure, and n representing engine speed. This expression can also be noted as follows because speed is constant

$$b_e \sim \frac{\dot{B}}{Pe}$$

This term indicates that given a certain fuel throughput ($\dot{B}$) it is only possible to reach a specific fuel consumption minimum if you reach a mean effective pressure maximum at the same time.

Consequently, whenever we have a case, as in Fig. 6.1.82, of the maximum moment and the minimum specific fuel consumption not coinciding in the same air/fuel ratio, fuel throughput must have been changed in between the two measurements. And this is indeed the case, for curves like the ones in Fig. 6.1.82 are generated by keeping the air throughput constant, which means keeping the throttle angle constant, and varying the fuel throughput.

Fig. 6.1.81 demonstrates that it is necessary to abandon this practice: The fat interrupted line connects points of minimum specific fuel consumption recorded at varying air throughput rates, whereas the fat uninterrupted line obviously connects points of minimum fuel through-

put at a given mean effective pressure. Taking into account the fine uninterrupted lines which represent air/fuel ratio, we can see from Fig. 6.1.81 that these points of minimum fuel throughput are associated with leaner mixtures than those obtained by varying fuel throughput at a constant air/fuel ratio.

Fig. 6.1.83 illustrates the aforesaid by showing test results obtained from a 1.3 l VW engine. The graph shows absolute fuel consumption over intake vacuum. The differences between the curves are due to their representing various points on the partial road load curve. They are basically lines of constant power output, along which the air/fuel ratio has been varied by changing both fuel and air throughput. If the engine speed is kept constant, the intake vacuum may be used to measure the air throughput. Of course, the best point on each line is the one in which absolute fuel consumption is lowest. Here again, however, this point unmistakably differs from that of minimum intake vacuum. That will be reached, however, if the throttle angle is kept constant and the mixture is made progressively leaner. The end product here would be a curve whose plane of reference is vertical relative to that of Fig. 6.1.83, that is why it is merely indicated by a straight line in that graph (see dotted line). Beginning at point A and running through progressively leaner mixtures, one would arrive at what would seem to be a point of minimum fuel consumption B, and then the curve would run up again to point C, the lean mixture limit.

Optimizing the air/fuel ratio along the full-load line differs from the ones named above. The goal here is to get the maximum possible torque from a given engine at any given speed. This kind of optimizing process has been tried on a VW 1.3 l engine as well (see Fig. 6.1.84). We selected the air/fuel ratios associated with the points of contact between the vertical tangents and the curves, because these points of contact indicate maximum torque and, therefore, maximum power output. As a rule, these points are not identical with the points of optimum fuel consumption.

All that has been said so far concerning air/fuel ratio and fuel consumption refers only to engines of ideal mixture distribution running in steady state and at operating temperature. The problem immediately becomes more complex as soon as the aspect of mixture distribution acceleration, deceleration, and warm-up phenomena are introduced.

As a matter of principle, mixture distribution should be as uniform as possible in all operating modes of an engine, because otherwise it will be impossible to adjust the engine to minimum fuel consumption. For the air/fuel ratio which is associated with minimum fuel consumption is so lean as to be close to the ignition failure point. If uniform mixture distribution cannot be guaranteed under these conditions, individual cylinders may well have ignition failures if the entire engine is adjusted to minimum fuel consumption.

Especially in carburetor engines the air/fuel ratio has to be enriched to get good driveability at the point of changeover from cruising to acceleration, although as a rule this is necessary in fuel injection engines as well. This modification tends to increase fuel consumption. Occasionally, it may be that certain exceptional engines do not follow this rule if they are tuned to a point below that of minimum fuel consumption to meet certain emission standards. One of the engines tested under this contract was of this kind: The 1.3 l engine, whose fuel consumption with the accelerating pump switched off was higher than with the pump switched on, as Fig. 6.1.85 shows.

The influence of the air/fuel ratio on fuel economy is most intensive during the cold start phase. To get the mixture in a cold engine to the point of ignition it has to be very rich, for only low-boiling fuel components will evaporate at low temperatures, and the percentage of the fuel which will evaporate decreases with the temperature. Now it is only evaporated fuel molecules well mixed with oxygen which will be ignited by the ignition spark, which, after all, is very small. Consequently, all engine technologies, no matter how sophisticated, have to make use of cold start enrichment. Still, there is a large number of ways to cut down on the time during which enrichment is necessary, and to cut down on the extent to which the mixture is enriched. One of these is to cut off the flow of coolant during cold starting and the phase immediately after, whereas another is to short-circuit the coolant circulation so that the coolant's capacity for heat absorption remains as low as possible.

Another group of devices runs under the heading of mixture preheating. This may involve either heating the fuel itself or the air/fuel mixture by cooling water, by exhaust gas, or by electric energy, all of which are applied as a rule to the intake manifold walls. A more recent development is to have electric PTC heaters inside the intake manifold itself.

Finally, there is the preheating of the intake air, which is done mainly by sucking the air away from walls preheated by exhaust gas, such as the walls of the exhaust manifold.

All this is done exclusively and only to keep both the extent and the duration of cold start enrichment as far down as possible, so that operating temperatures and optimum air/fuel ratios may come into play as soon as possible.

It was especially under the impression made by the future U.S. exhaust emission regulations whose CO and HC standards can be met only by improving the cold start phase that strenuous efforts were made in this field, and it is possible now to cut the cold phase in the U.S. exhaust emission test down to 60 seconds.

The actual extent to which cold start enrichment influences the fuel consumption of an automobile in practice depends of course on the average distance covered between two cold starts. The shorter this distance, the higher is the influence of cold start enrichment. From a certain average daily mileage on up, however, the influence of cold start enrichment becomes indistinct. The results of the tests performed by us in this context are represented by the scatter band in Fig. 6.1.86.

Ignition Timing

Proper ignition timing is another factor on which the fuel economy of an engine depends. If ignition is advanced too far, much of the pressure increase which is caused by combustion occurs before TDC and the movement of the piston is impeded, which means negative work. If, on the other hand, ignition is delayed too much, energy conversion largely takes place on a very low thermal level, so that the heat of the combustion process is not exploited optimally. Yet, in spite of all this, there is nothing at the moment which would require extremely precise ignition timing.

Engines under full load are relatively sensitive to undue ignition advancement. Fig. 6.1.87 shows engine power output over ignition angle at various engine speeds and at full throttle. As all engine speeds given here are associated with maximum air flow and an optimum air/fuel ratio geared for maximum power output, the performance curve may be assumed to be indirectly proportional to the specific fuel consumption. Consequently, the points of maximum performance on these curves coincide with points of minimum specific fuel consumption.

Fig. 6.1.88 uses the data from the previous figure to indicate the optimum advance angle at each engine speed as well as the extent of the advance range within which performance at full throttle will deteriorate by no more than 1 %.

The width of this range fluctuates between about 5 and 10° crank angle, its fluctuations obviously depending mainly on the variations of the air/fuel ratio along the full load curve. Variations of 5° must be allowed because it is impossible to ensure adherence to lesser ignition timing tolerances in production and in the field with the ignition systems which are still in general use today. Even a variation of 10° is still sufficient to obtain near-optimum performance and optimum fuel consumption. However, as we can see in Figs. 6.1.87 and 6.1.88 the knock limit may present some difficulties here, because at low engine speeds the optimum ignition timing is already within the range of knock.

Figs. 6.1.89 and 6.1.90 present a similar situation referring to various engine speeds along the partial road load curve. It is especially the last-named figure which shows that engines under partial load are no more than half as sensitive to ignition timing changes as they are under full load, where their sensitiveness is none too excessive in the first place.

To sum up, we may say that it is questionable whether overly exact ignition timing, at whatever point on the engine map, does have any effect whatsoever on fuel economy. All we can do in the field of ignition timing today is to have an ignition system which over the entire life of an engine will keep to a timing as good as that of conventional systems is on leaving the factory.

The question whether it is possible to influence combustion after the ignition proper has taken place is another matter altogether. Here, the initial stage of combustion deserves special attention, i.e. the time which elapses from the extinction of the ignition spark until 1 % of the charge is burned. Once we succeed in gaining an understanding of this process sufficient to control it, precise ignition timing will certainly be of use in attaining considerable improvements in fuel consumption.

EGR

If without changing any of the other adjustments exhaust gas is recirculated into an engine adjusted to minimum fuel consumption without changing, the fuel economy of this engine will deteriorate as a general rule. However, this is by no means the whole extent of the problem, because the relationship between fuel economy and EGR is far more complex than may seem at first glance.

If, for instance, we vary the quantity of exhaust gas recirculated and the ignition angle simultaneously we find that EGR may be of some help in improving fuel economy.

Fig. 6.1.91 (12) is a presentation of a relationship of this kind. Starting with a certain operating mode defined by throttle angle, ignition time, and air/fuel ratio, we varied the ignition timing, which resulted in the curve running through the point of origin which is identified by the legend '0 % EGR', being at 100 % NOx, 100 % fuel consumption, and 100 % mean effective pressure. If EGR is adjusted to 5 % of the total air intake, the emission of NOx drops to about 80 % without any change in fuel consumption. Further increases in the EGR percentage will eventually, at about 15 % and more, produce an increase in fuel consumption.

If, however, EGR is kept at 15 % and the ignition is advanced to about 40°, the quantity of NOx emitted will increase again from 37 to 60 % or thereabouts, but this is accompanied by a 3 % improvement in fuel economy. The graph shows that it is possible to improve both fuel consumption and mean effective pressure by about 5 % each by increasing the EGR percentage to 25 and advancing the ignition by 50° or so.

There is still, however, the question whether the relationship shown in this figure represents a general rule or whether it is merely the result of a more or less random coincidence of parameters. The answer to this question follows from the line of reasoning given below.

In an ideal internal combustion engine, efficiency cannot be positively affected by exhaust gas recirculation, the efficiency of an ideal engine being described by the following equation:

$$\eta = 1 - \frac{1}{\varepsilon^{\varkappa - 1}}$$

In this equation, ε represents the compression ratio, whereas $\varkappa$ represents the adiabatic exponent of the process gas. The compression ratio remains unaffected by EGR. The adiabatic exponent, however, decreases as soon as EGR is introduced because of the loss of bi-atomic gas whose adiabatic exponent is 1.4. Therefore, the result is that the efficiency of the ideal engine deteriorates.

Another reason why the efficiency of the ideal engine is deteriorated by EGR is that EGR delays combustion. The ideal charge cycle presumes that the heat of combustion is generated and absorbed in zero time. Therefore, EGR is bound to entail a loss of efficiency since it extends that period of time.

However, these two factors are counteracted by the fact that the introduction of EGR necessitates opening the throttle somewhat more if the power output of the engine is to remain constant, which means a reduction in charge cycle losses and an increase in final compression pressure. Still, even a rough estimate will show that this positive influence is more than belanced by the two negative influences, especially so as it is associated with increased ventilation losses occurring in the EGR piping and in the valves. In other words: In an optimal engine, we have to assume that its fuel economy will as a rule be deteriorated by the introduction of EGR. And yet there are the test results shown in Fig. 6.1.91, which are beyond any doubt correct.

This contradiction between our test results and our line of reasoning about the influence of exhaust gas recirculation on the efficiency of an engine can be resolved by adopting the theoretical construction in Fig. 6.1.92 which presumes that the mean effective pressure remains constant. That this is so in every case is effected by varying the throttle angle. The graph indicates fuel consumption in time over air/fuel ratio. Let us assume that the uninterrupted curves represent a spark advance of 40°, the interrupted curves an advance of 20°, and the dotted curves a spark advance of 60°.

Under assumption of a constant mean effective pressure the lowest fuel consumption would occur at a lean λ of more than 1, at a spark advance of 40°, and at 0 % EGR. This point, point 1, is therefore the point of optimum operation. If mean effective pressure, spark advance, and air/fuel ratio are kept constant while, for instance, 5 % EGR is introduced the throttle must be opened to point 2. Consequently, EGR has brought about a loss of efficiency.

If spark advance, mean effective pressure, and EGR are kept constant while the operating point of the engine is slowly shifted towards richer air/fuel ratios there should be an improvement in fuel economy, because somewhere along the line, you are bound to approach another point of minimum fuel consumption (3). The mixture at this point is bound to be richer than at minimum 1 because both EGR and a lean mixture tend to delay combustion.

This is also the reason why there have to be points on the engine map where EGR initiates an improvement in fuel economy. These are the points where at a given spark advance and a given air/fuel ratio combustion without EGR would be so fast that it would largely impede the movement of the pistons. Here, therefore, EGR lowers the speed of combustion, thus contributing towards improving fuel economy.

There is a point of intersection (4) between the shaded area, where fuel consumption is improved by exhaust gas recirculation, and the dotted area, where it is adversely affected. At this point, given a constant air/fuel ratio, mean effective pressure, and spark advance, fuel consumption is the same with and without EGR. A point like this must exist because there must be a certain air/fuel ratio where, all other conditions being the same. The deterioration of the adiabatic exponent which is caused by EGR is just balanced by the improvement in the charge cycle and the improvement due to the EGR caused combustion delay.

The interrupted curves are another presentation of the same relationship at a lesser spark advance.

It is easy to see that under these conditions it is entirely possible to obtain test results such as those shown in Fig. 6.1.93, where the efficiency is actually improved by EGR. However, these results can be properly understood only if one is aware that the original operating point was one in which throttle angle, spark advance, and air/fuel ratio were not optimal.

Fig. 6.1.92 is a mere theoretical construction. We have to leave it to the future to see if it can be corroborated by test results.

Given the state of the art of today, fuel economy is in most cases adversely affected by exhaust gas recirculation.

Fig. 6.1.93 illustrates this by giving the results of dynamic tests run under this Contract according to the requirements of the '72 Federal Test Procedure. The figure marked '26' is an average drawn from five measurements performed with constant adjustments. The engine corresponds to Modification Code 17 (see Chapter 4.2.1).

Our main goal was to improve fuel economy and NOx emissions. The first step (A) in this investigation is test 44, which is run with the spark retard diaphragm deactivated, so that, in the range close to idling speed, there is a spark advance of 7.5° CA instead of a spark retard from 3° CA. The fuel economy improved slightly, which is according to expectations. The next step (B) was to keep the spark retard diaphragm deactivated while reducing the basic air/fuel ratio of the K-Jetronic injection system to a normal level of leanness by increasing the control pressure from 3.0 to 3.55 bar. This brought about a further improvement in fuel economy, but the emissions of NOx rose to 0.3 gpm.

In the next test (47), step C, we determined the influence of an EGR system (see Fig. 4.2.9) governed by the engine air throughput (see Chapter 4.2.1 modification code 18). We found that in this way we were able to remain below the engineering goal of 0.1 gpm NOx, at the cost, however, of a fuel consumption which was higher by nearly 10 % compared to the average of all tests run on the engine adjusted as in test 26. By advancing the spark time throughout the entire engine map by 5°, which was done in the next test (49), step D, we were again able to produce a marked improvement in fuel economy at the cost of a minimal deterioration in NOx emissions.

In Fig. 6.1.91, this would correspond to moving from point A along the constant spark advance line to point B and then along the line of constant EGR to point C, in Fig. 6.1.92, it would approximately mean moving from point 5 to point 7 via point 6.

Intake Air Temperature

We are not aware of any tests having been run to determine the direct influence of the air intake temperature on fuel consumption. However, it is a fact that increasing the intake air temperature leads to a lesser charge and to an increase in friction, which again influences fuel economy adversely. Moreover, high intake air temperatures mean unfavorable dynamic processes and thus a loss of internal efficiency too.

In addition to this, the intake air temperature may be an influential factor in some indirect ways: If it increases, the temperature of the charge will increase as well, and at WOT this will make the engine more prone to knock (see Fig. 6.1.57 in Chapter 6.1.10). For this reason, undue increases in the intake air temperature should be avoided. On the other hand, it may be imperative to have a high intake air temperature in order to ensure good mixture distribution. In that case, compression would have to be reduced, and this would lead to more fuel consumption.

In the partial load range of the engine map, which is relatively wide, the intake air temperature may be increased almost at will, because it helps to improve mixture formation and mixture distribution. In this range, high intake air temperatures tend to improve fuel economy as it is partly due to them that engines can be adjusted to mixtures lean enough to permit minimum specific fuel consumption.

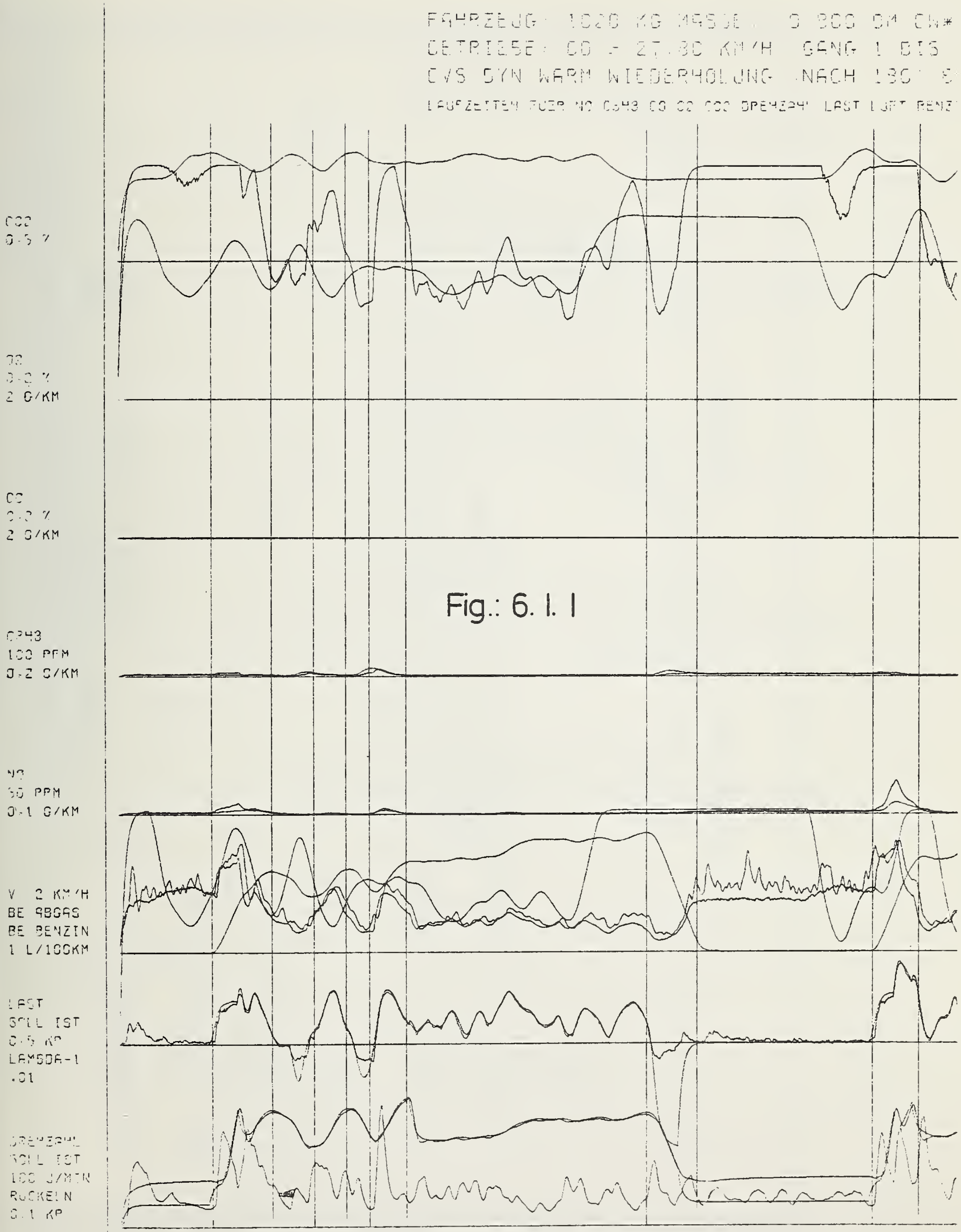


Fig.: 6. 1. 1

PCS-CONTROLLED TEST DRIVE CYCLE TEST RECORDS

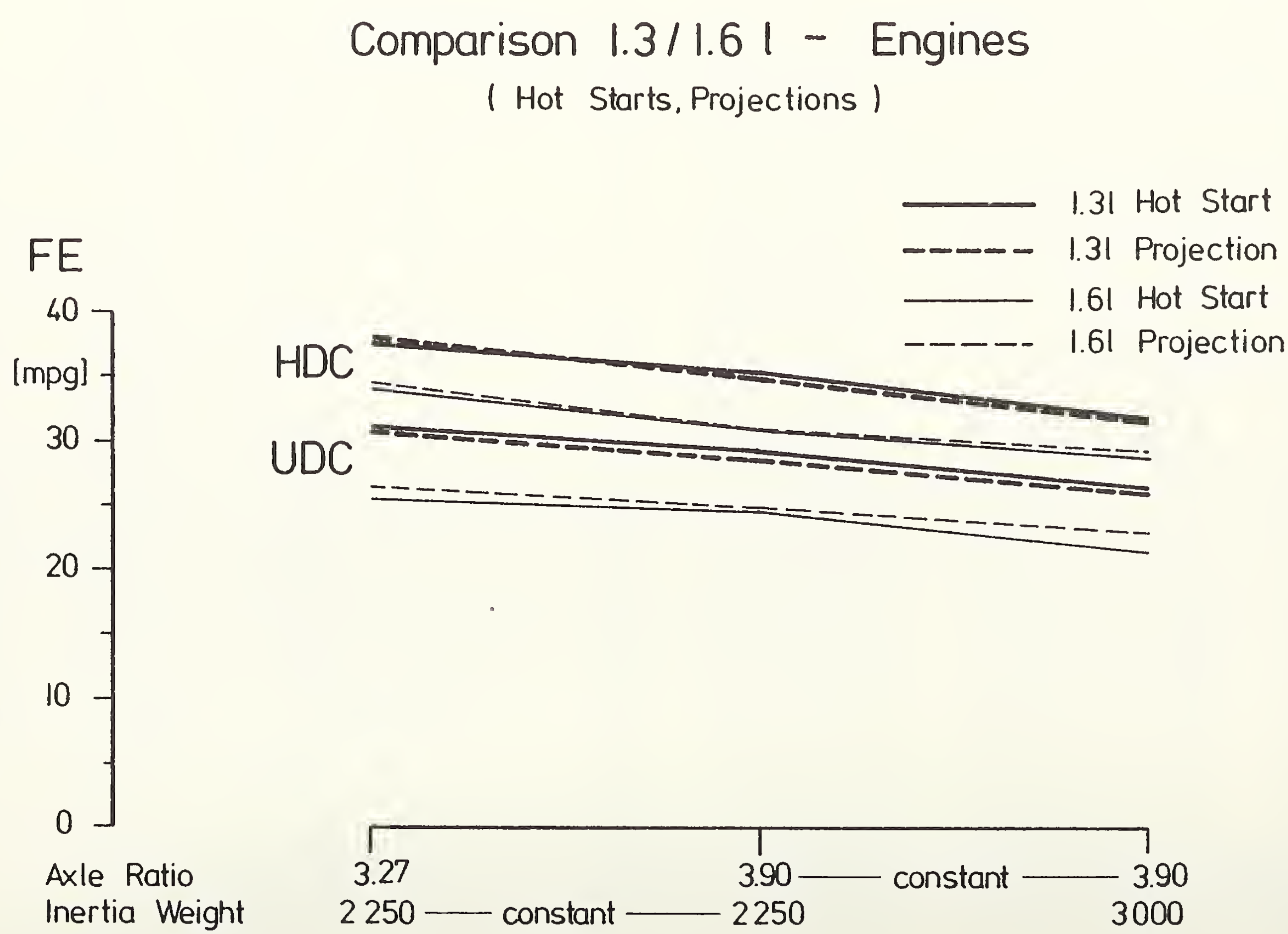
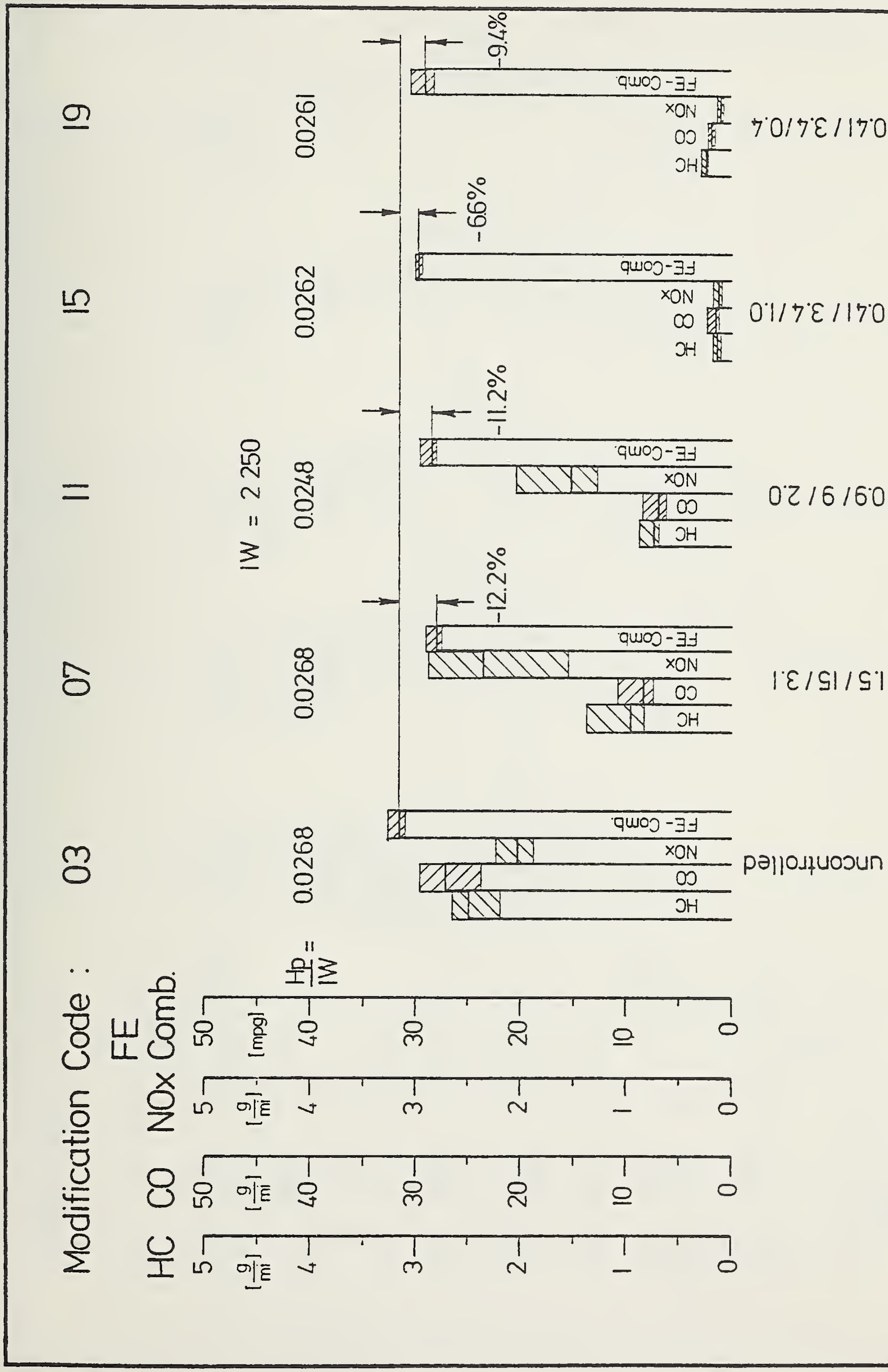

Comparison 1.3 / 1.6 l - Engines
(Hot Starts, Projections)
1.3l Hot Start
1.3l Projection
1.6l Hot Start
1.6l Projection
FE
40
[mpg]
30
20
10
0
HDC
UDC
Axle Ratio
Inertia Weight
3.27
2 250 constant 2 250
3.90 constant 3.90
3000

Fig.: 6. 1. 2



Modification Code :
03
07
11
15
19
HC CO NOx FE Comb.
Hp/IW =
0.0268
0.0268
0.0248
0.0262
0.0261
IW = 2 250
[g/mi]
[g/mi]
[g/mi]
[mpg]
-12.2%
-11.2%
-6.6%
-9.4%
HC
CO
NOx
FE - Comb.
uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 6. 1. 3

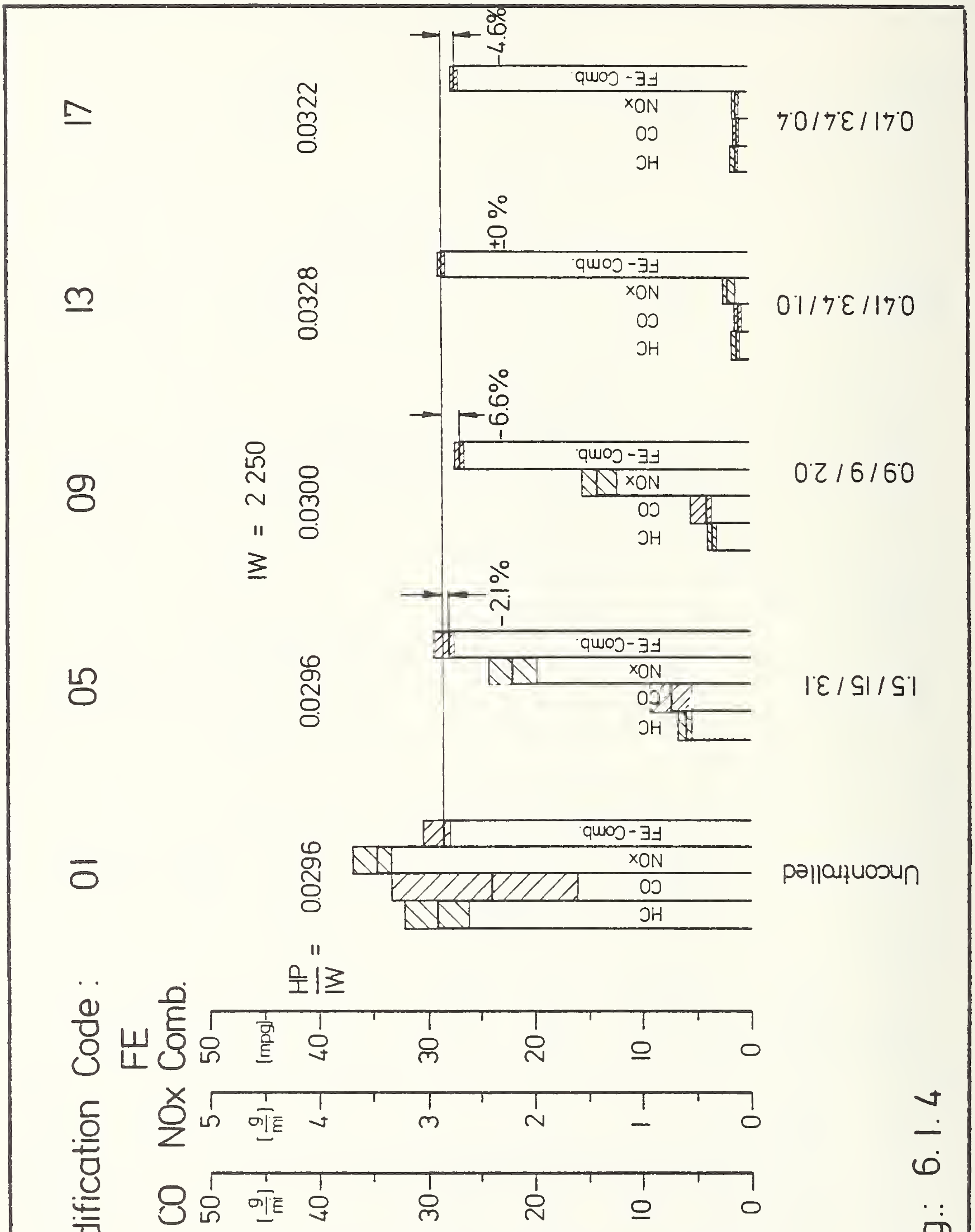

Modification Code :
01
05
09
13
17
HP / IW =
0.0296
0.0296
0.0300
0.0328
0.0322
IW = 2 250
HC [g/mi]
CO [g/mi]
NOx [g/mi]
FE Comb. [mpg]
HC
CO
NOx
FE - Comb.
-2.1%
-6.6%
±0 %
-4.6%
Uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 6.1.4

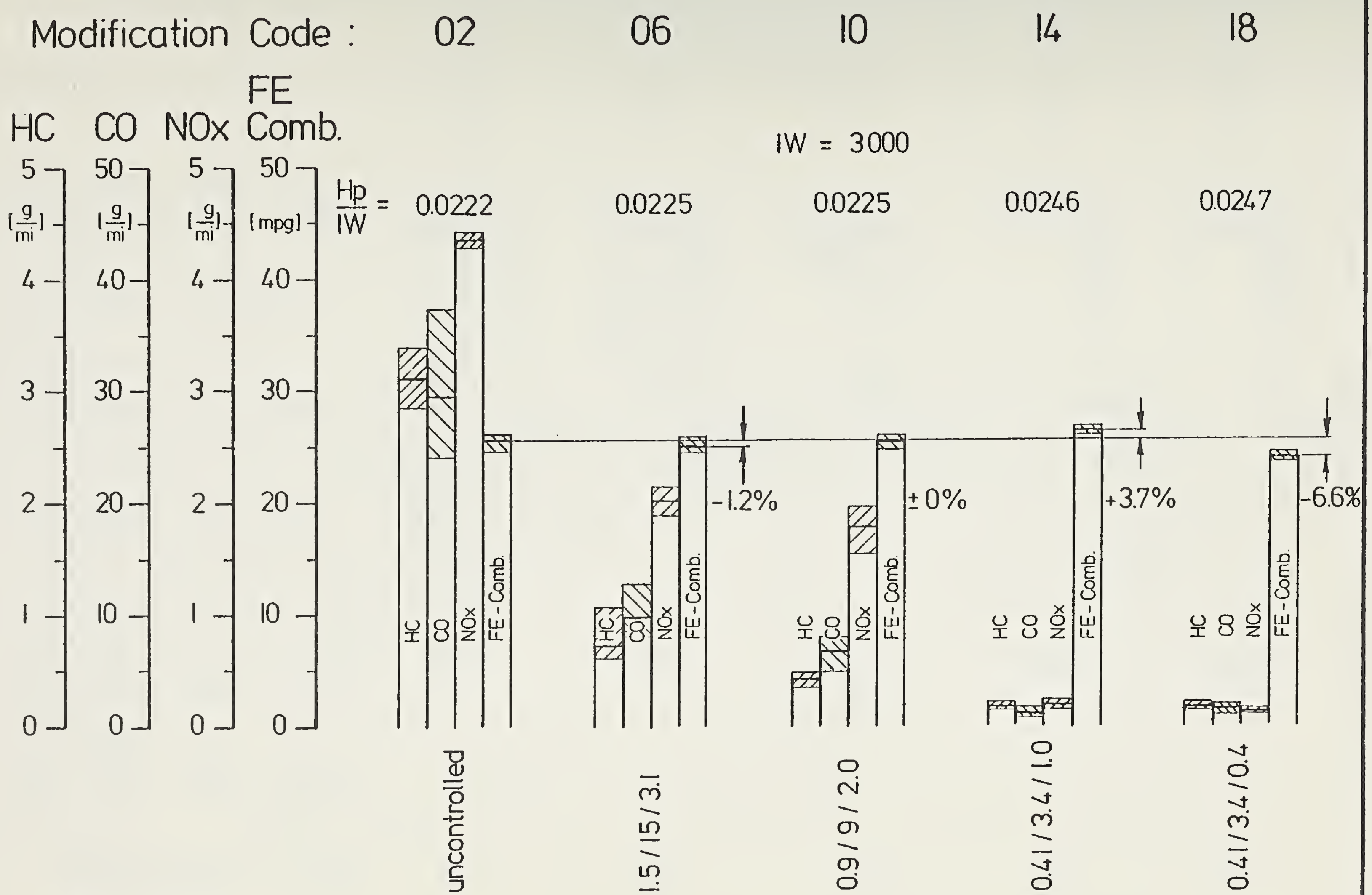

Modification Code :
02
06
10
14
18
HC
CO
NOx
FE Comb.
[g/mi]
[g/mi]
[g/mi]
[mpg]
IW = 3000
Hp/IW =
0.0222
0.0225
0.0225
0.0246
0.0247
HC
CO
NOx
FE-Comb.
-1.2%
±0%
+3.7%
-6.6%
uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 6. I. 5

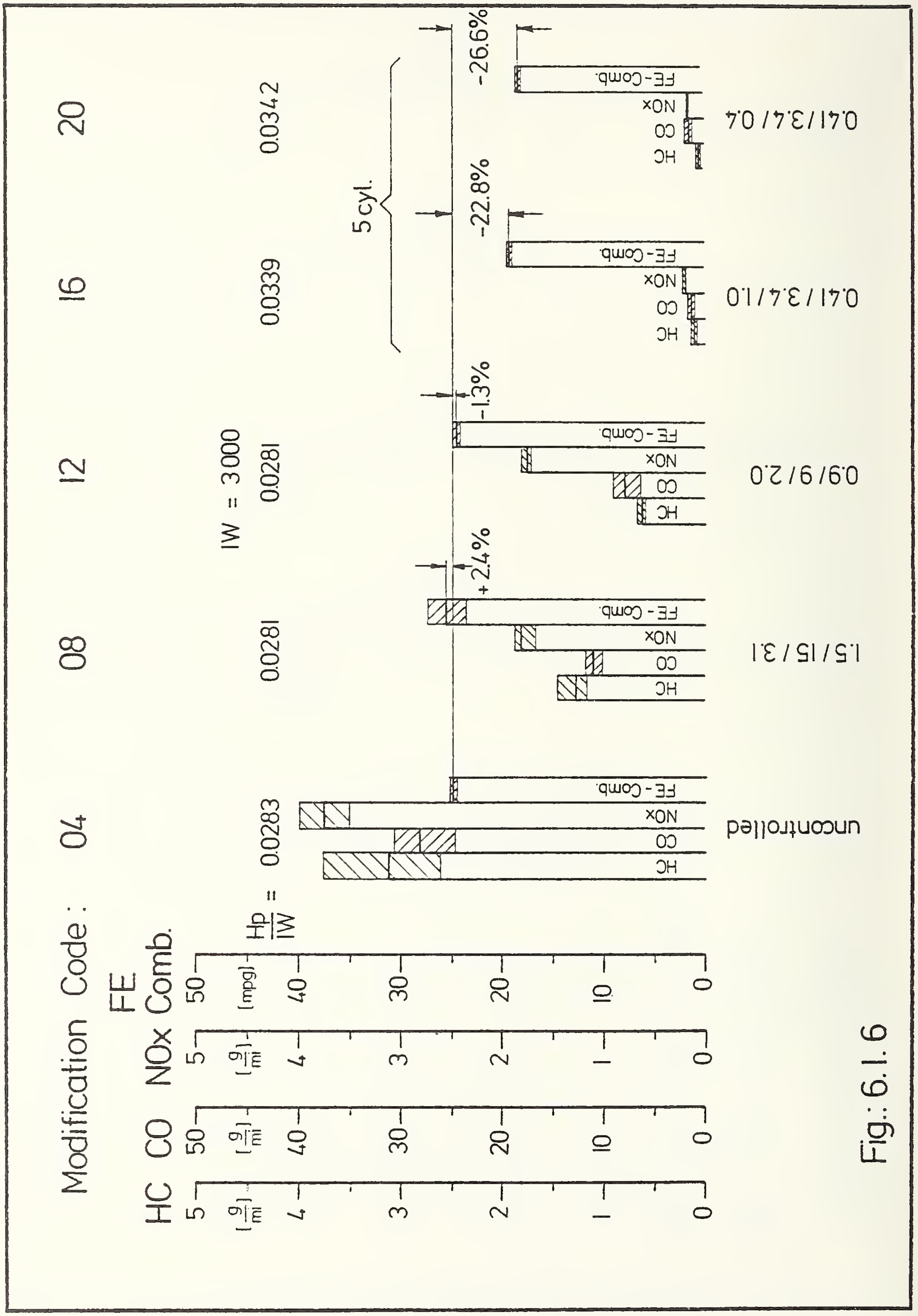
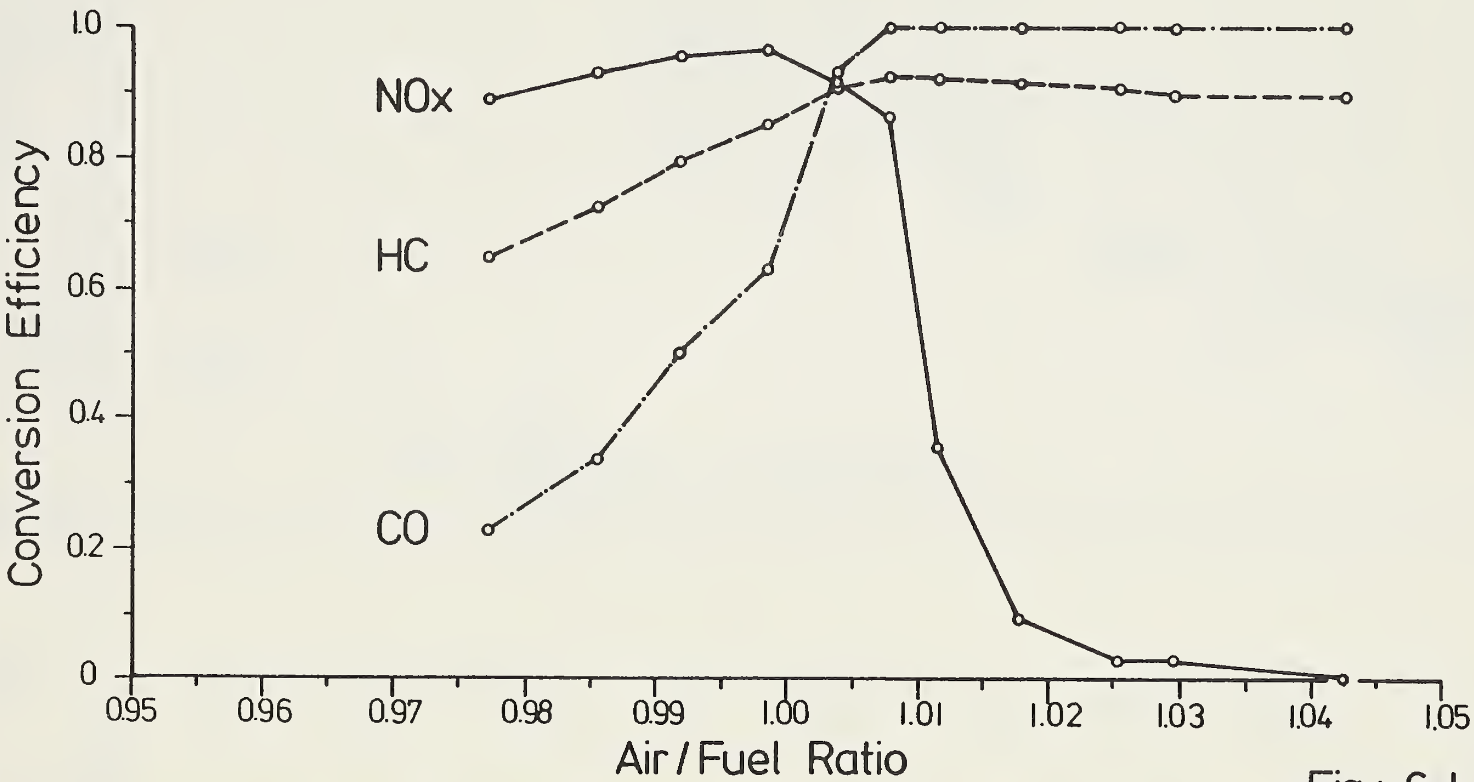

Modification Code :
04
08
12
16
20
Hp/IW =
0.0283
0.0281
0.0281
0.0339
0.0342
IW = 3000
5 cyl.
HC
CO
NOx
FE Comb.
[g/mi]
[g/mi]
[g/mi]
[mpg]
+2.4%
-1.3%
-22.8%
-26.6%
HC
CO
NOx
FE - Comb.
uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 6.1.6



Conversion Efficiency
1.0
0.8
0.6
0.4
0.2
0
0.95
0.96
0.97
0.98
0.99
1.00
1.01
1.02
1.03
1.04
1.05
Air / Fuel Ratio
NOx
HC
CO

Fig.: 6. I. 7

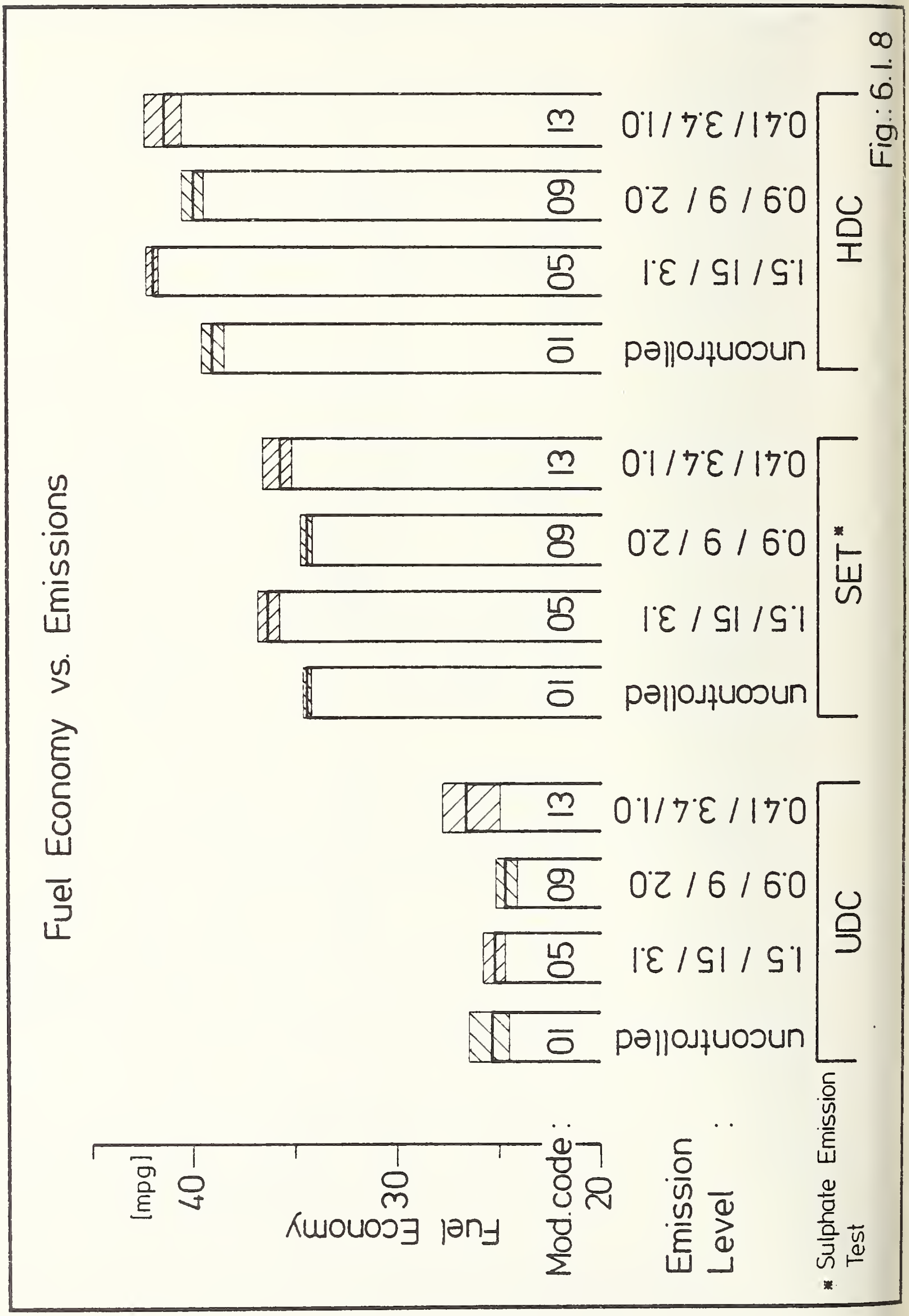

Fuel Economy vs. Emissions
[mpg]
40
30
20
Fuel Economy
Mod.code :
01
05
09
13
Emission Level :
uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
UDC
SET*
HDC
* Sulphate Emission Test

Fig.: 6.1.8

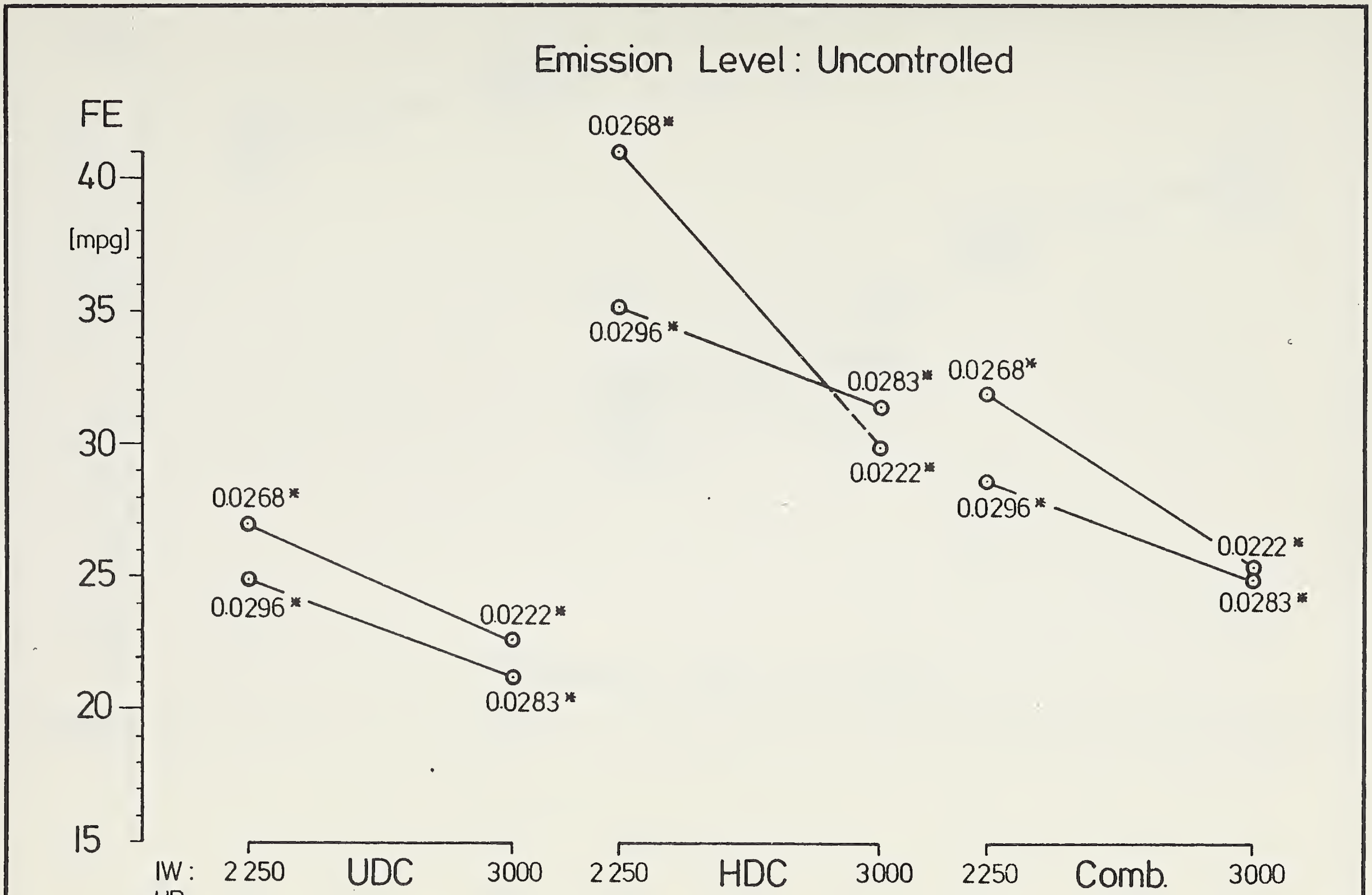

Emission Level : Uncontrolled
FE
[mpg]
40
35
30
25
20
15
0.0268 *
0.0296 *
0.0222 *
0.0283 *
0.0268 *
0.0296 *
0.0283 *
0.0222 *
0.0268 *
0.0296 *
0.0222 *
0.0283 *
IW :
2 250
UDC
3000
2 250
HDC
3000
2250
Comb.
3000
* = HP/IW

Fig.: 6.1.9

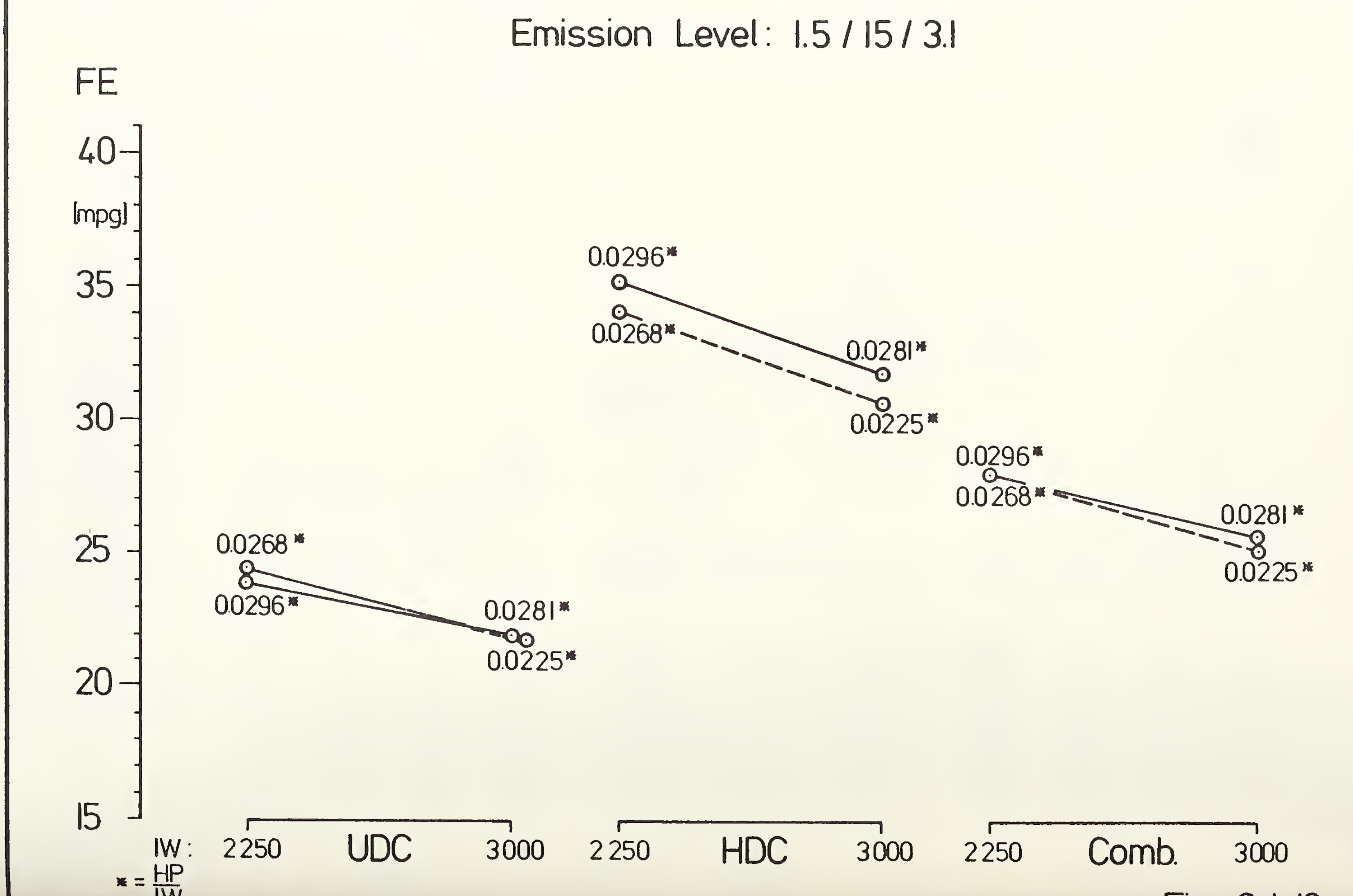

Emission Level: 1.5 / 15 / 3.1
FE
[mpg]
40
35
30
25
20
15
0.0268*
0.0296*
0.0281*
0.0225*
0.0296*
0.0268*
0.0281*
0.0225*
0.0296*
0.0268*
0.0281*
0.0225*
IW:
2250
UDC
3000
2250
HDC
3000
2250
Comb.
3000
* = HP/IW

Fig.: 6.1.10

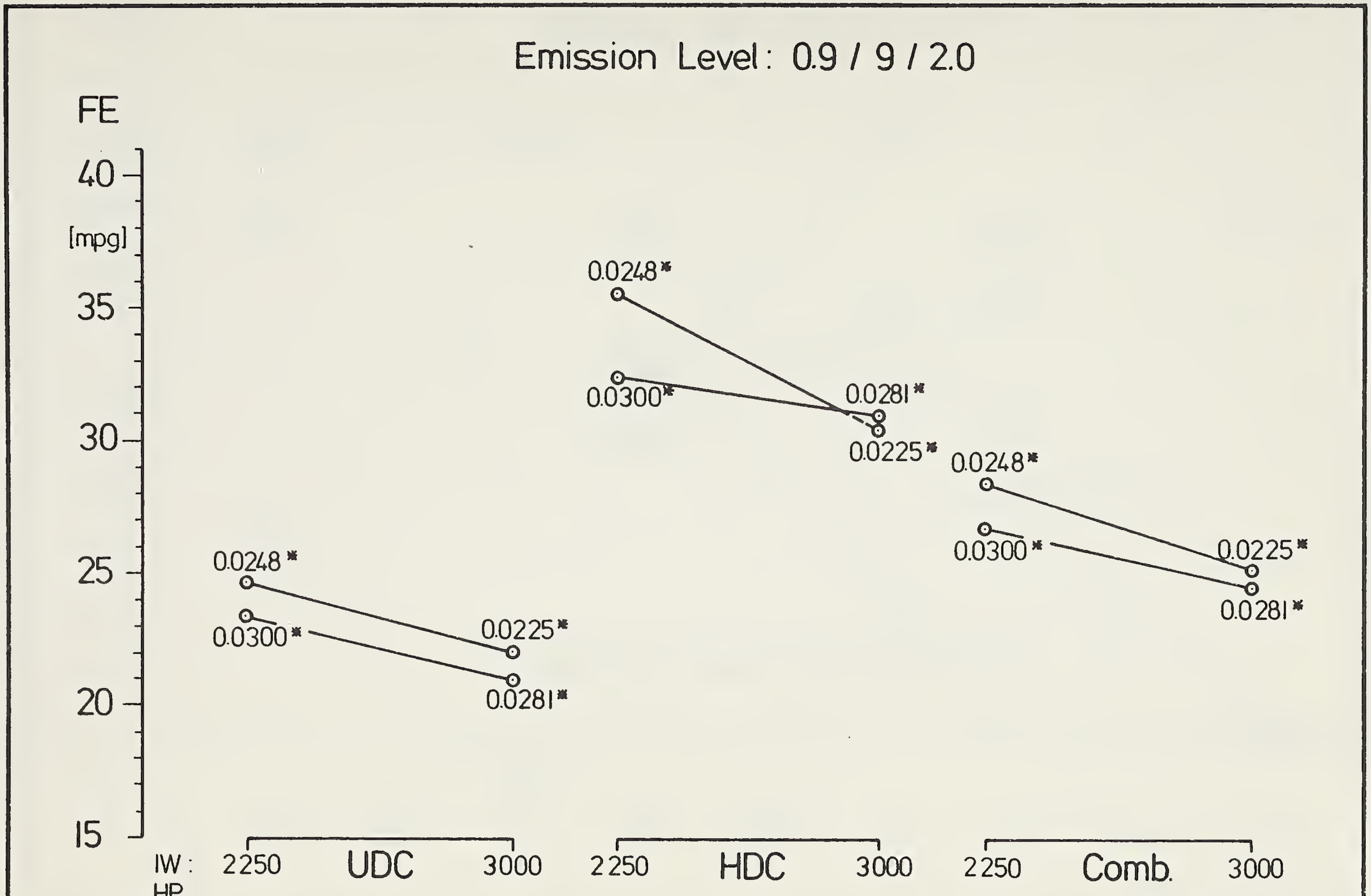
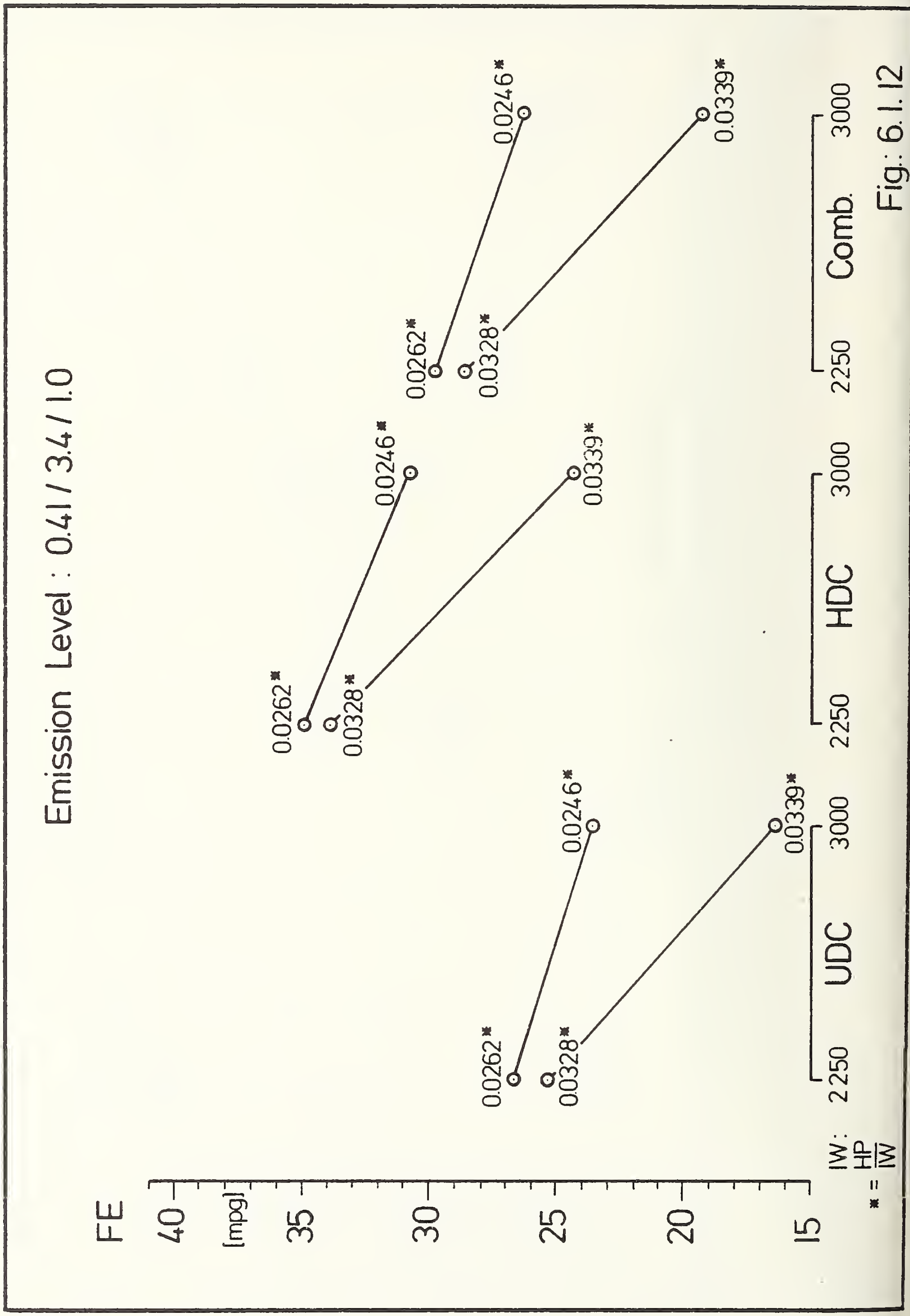

Emission Level: 0.9 / 9 / 2.0
FE
[mpg]
40
35
30
25
20
15
0.0248*
0.0300*
0.0225*
0.0281*
0.0248*
0.0300*
0.0281*
0.0225*
0.0248*
0.0300*
0.0225*
0.0281*
IW:
2250
UDC
3000
2250
HDC
3000
2250
Comb.
3000
* = HP/IW

Fig.: 6.1.11



Emission Level : 0.41 / 3.4 / 1.0
FE
[mpg]
40
35
30
25
20
15
IW:
* = HP/IW
UDC
2250
3000
HDC
2250
3000
Comb.
2250
3000
0.0262*
0.0328*
0.0246*
0.0339*
0.0262*
0.0328*
0.0246*
0.0339*
0.0262*
0.0328*
0.0246*
0.0339*

Fig.: 6.1.12

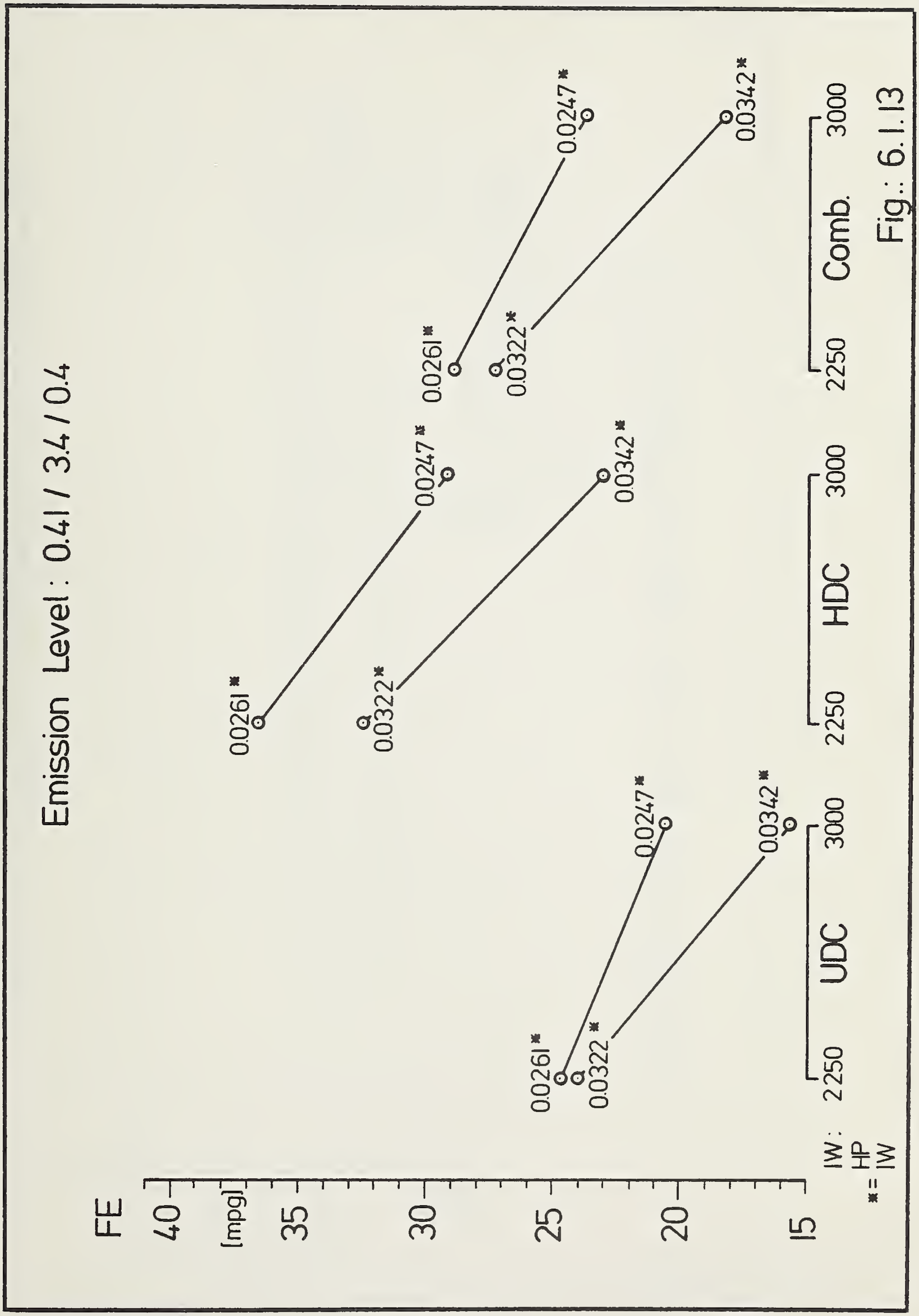

Emission Level : 0.41 / 3.4 / 0.4
FE
[mpg]
40
35
30
25
20
15
IW :
HP
* = IW
2250
3000
UDC
HDC
Comb.
0.0261*
0.0322*
0.0247*
0.0342*

Fig.: 6.1.13

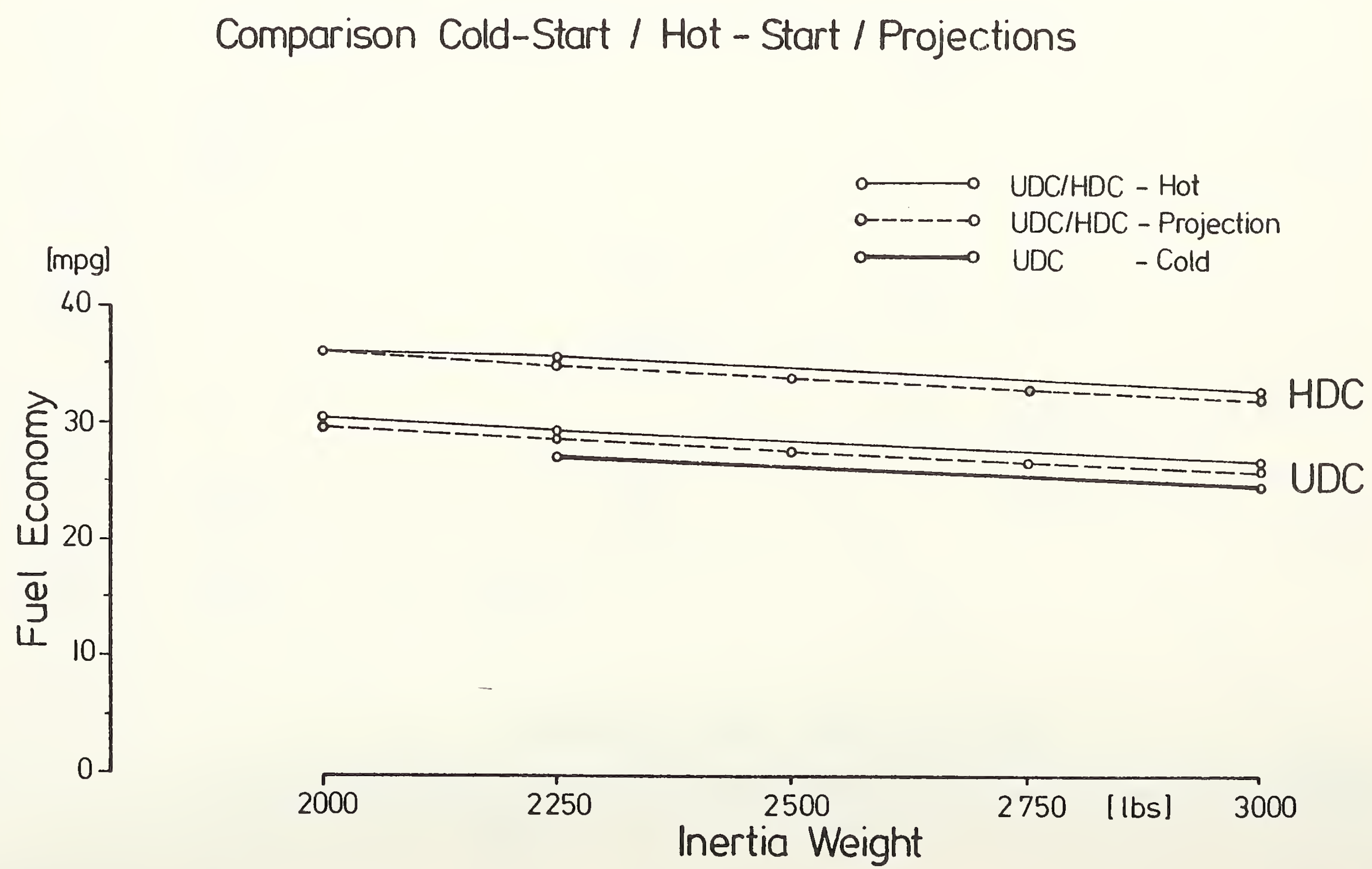
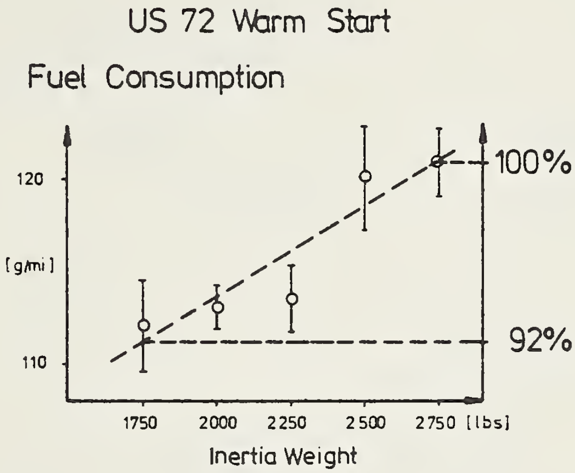

Comparison Cold-Start / Hot - Start / Projections
UDC/HDC - Hot
UDC/HDC - Projection
UDC - Cold
[mpg]
Fuel Economy
40
30
20
10
0
HDC
UDC
2000
2250
2500
2750
[lbs]
3000
Inertia Weight

Fig.: 6. 1. 14



US 72 Warm Start
Fuel Consumption
120
[g/mi]
110
100%
92%
1750
2000
2250
2500
2750 [lbs]
Inertia Weight

Fig.: 6. 1. 15 [1]

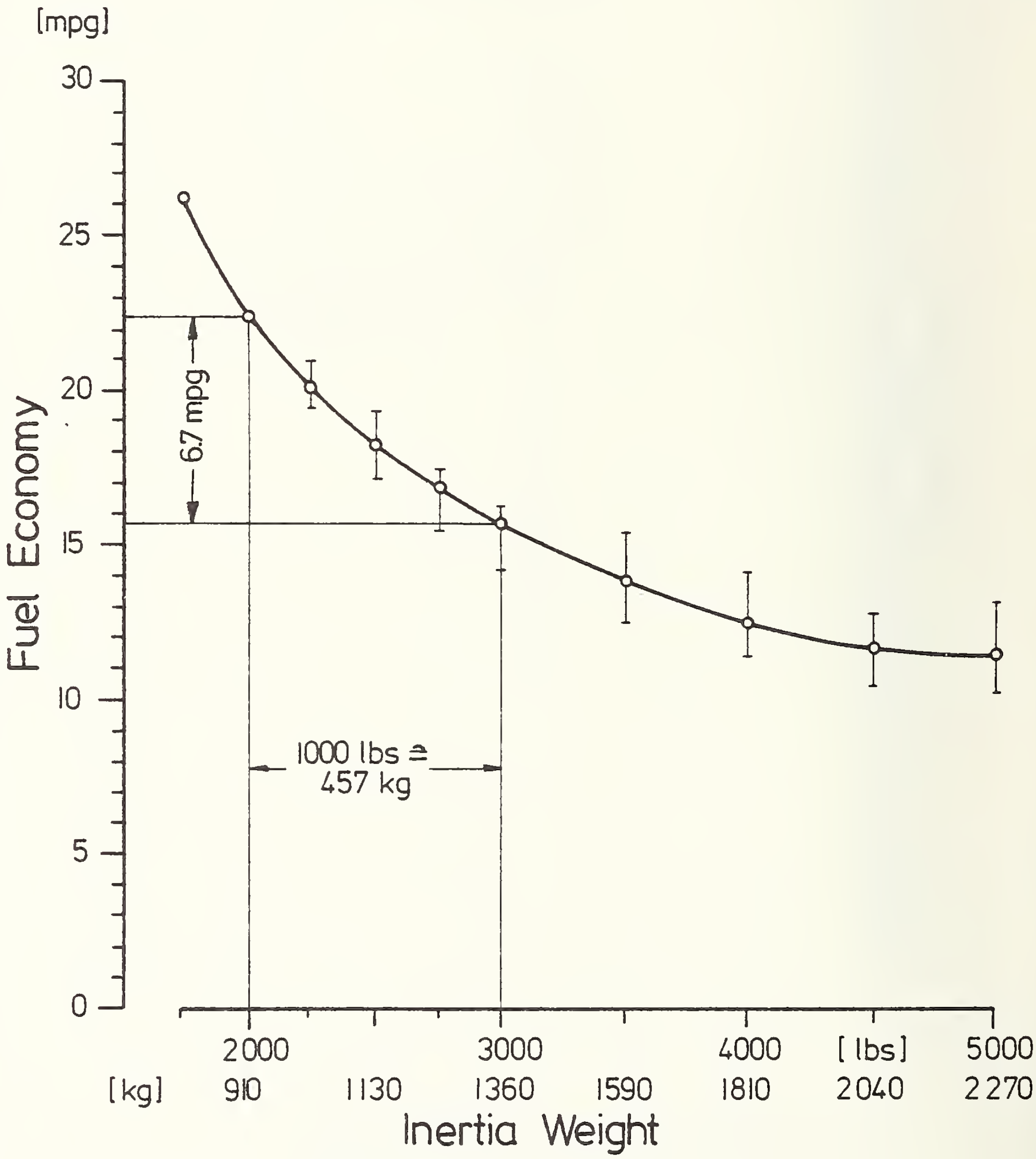
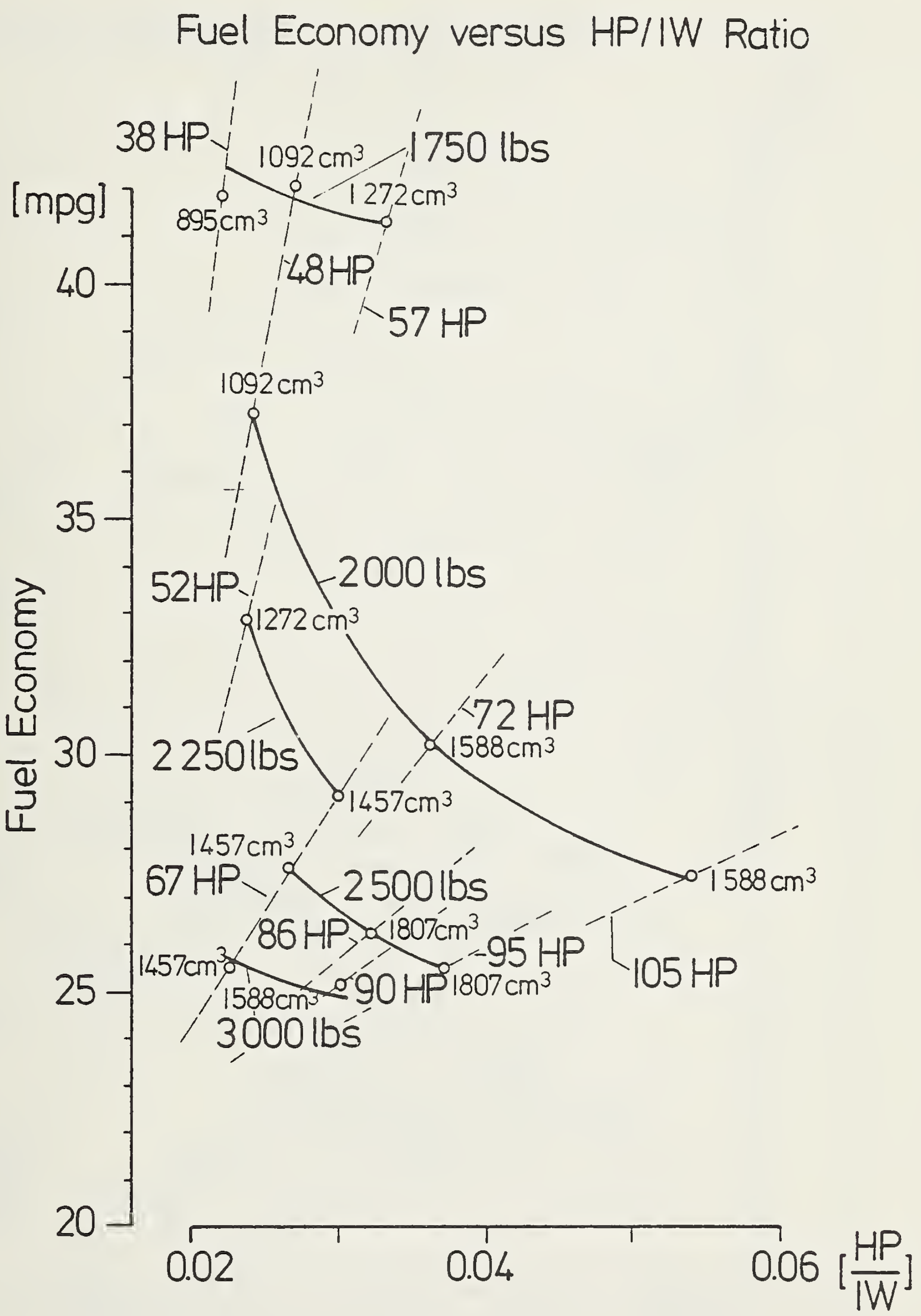
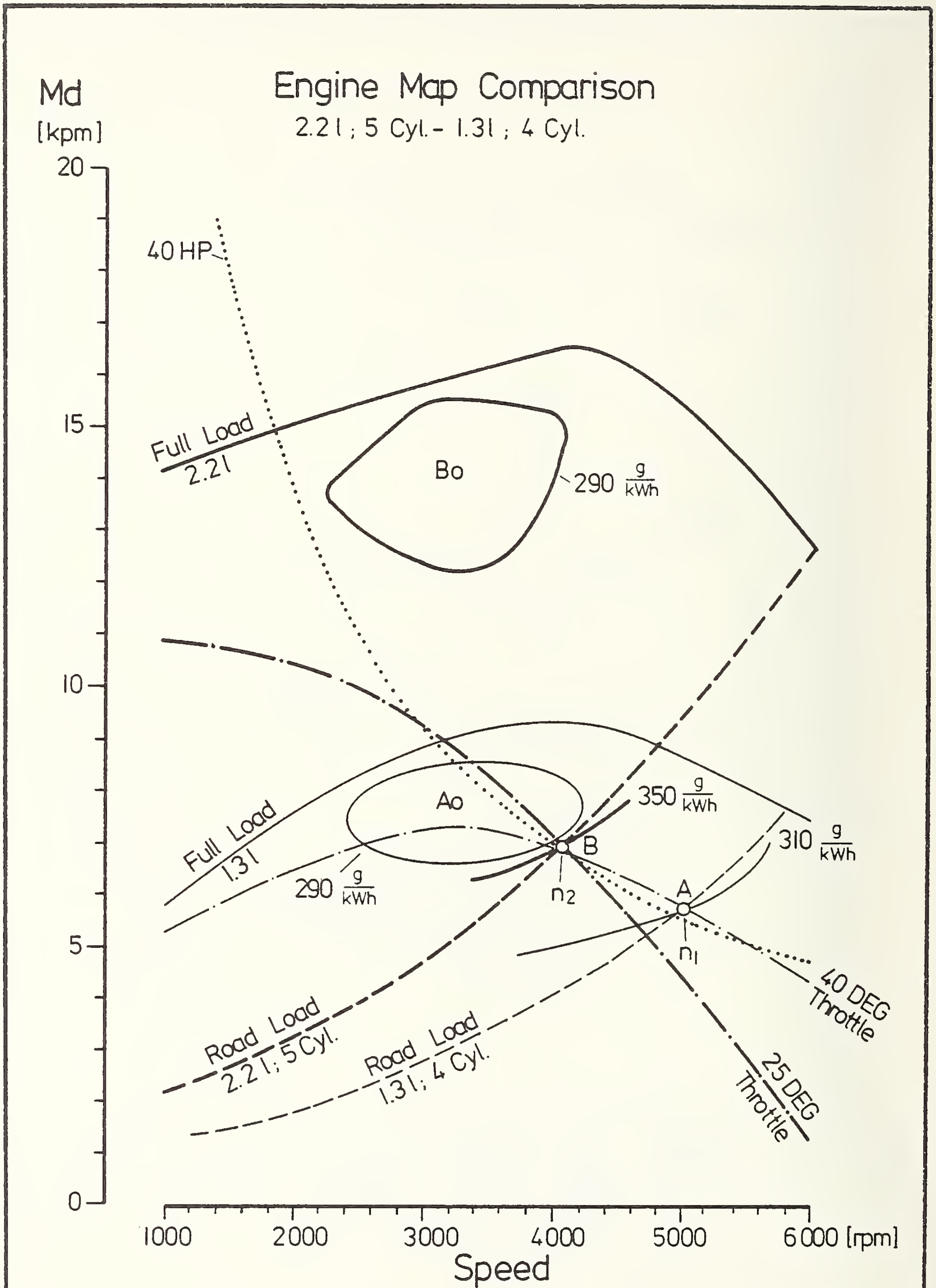

[mpg]
30
25
20
15
10
5
0
Fuel Economy
6.7 mpg
1000 lbs ≙
457 kg
2000
3000
4000
[lbs]
5000
[kg]
910
1130
1360
1590
1810
2040
2270
Inertia Weight

Fig.: 6. 1. 16



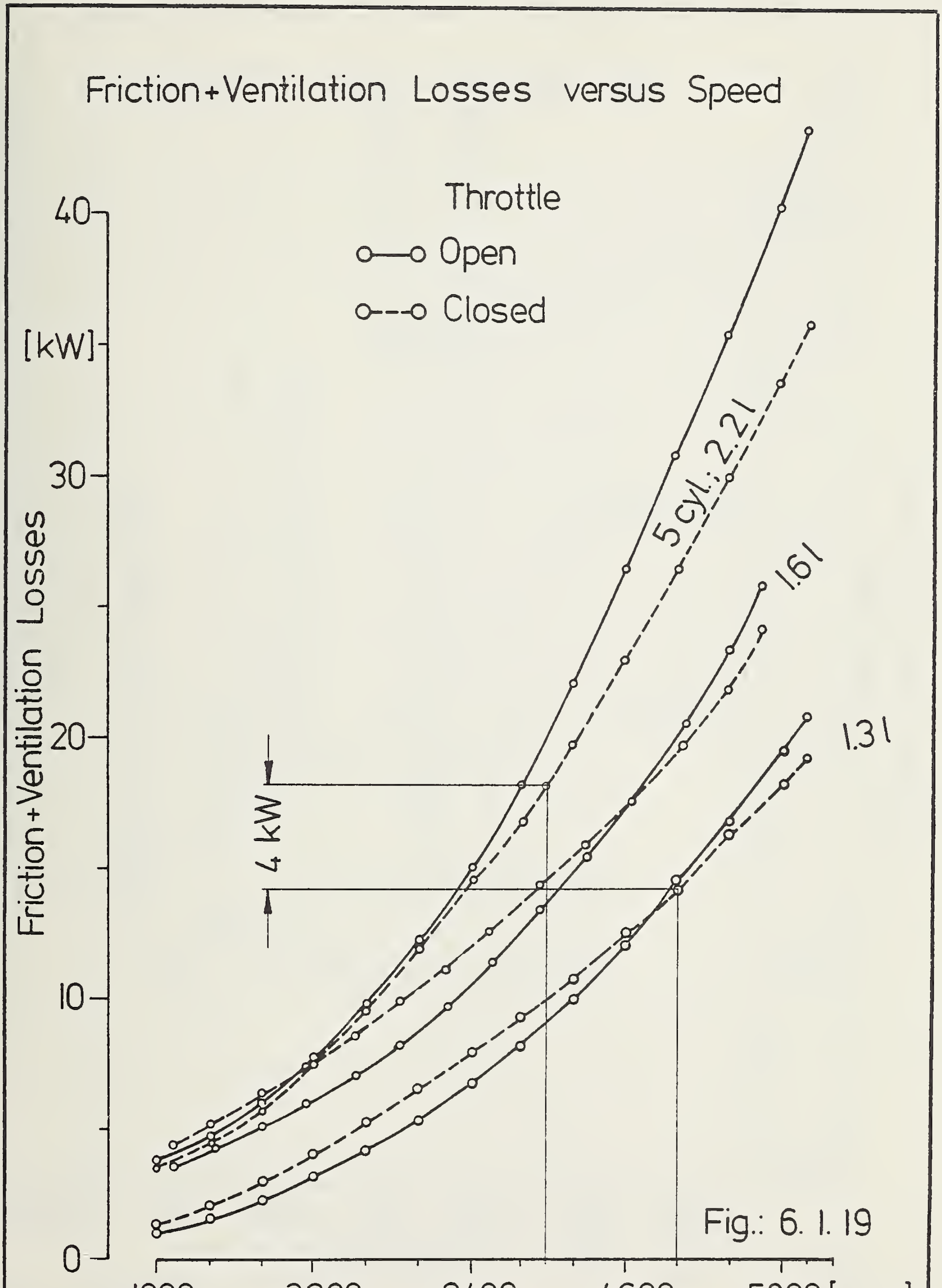
Fuel Economy versus HP/IW Ratio
[mpg]
Fuel Economy
40
35
30
25
20
0.02
0.04
0.06
[HP/IW]
38 HP
1092 cm3
1750 lbs
1 272 cm3
895 cm3
48 HP
57 HP
1092 cm3
2000 lbs
52HP
1272 cm3
72 HP
1588cm3
2 250 lbs
1457cm3
1457cm3
67 HP
2 500 lbs
1 588 cm3
86 HP
1807cm3
95 HP
105 HP
1457cm3
1588cm3
90 HP
1807 cm3
3 000 lbs

Fig.: 6. 1. 17



Md
[kpm]
Engine Map Comparison
2.2 l ; 5 Cyl. - 1.3 l ; 4 Cyl.
20
15
10
5
0
40 HP
Full Load
2.2 l
Bo
290 g/kWh
Ao
350 g/kWh
310 g/kWh
Full Load
1.3 l
290 g/kWh
B
n2
A
n1
40 DEG
Throttle
25 DEG
Throttle
Road Load
2.2 l; 5 Cyl.
Road Load
1.3 l; 4 Cyl.
1000
2000
3000
4000
5000
6000 [rpm]
Speed

Fig.: 6. 1. 18



Friction+Ventilation Losses versus Speed
Throttle
Open
Closed
40
[kW]
30
20
10
0
Friction+Ventilation Losses
5 cyl.: 2.2 l
1.6 l
1.3 l
4 kW
1000
2200
3400
4600
5800
[rpm]
Speed

Fig.: 6. 1. 19

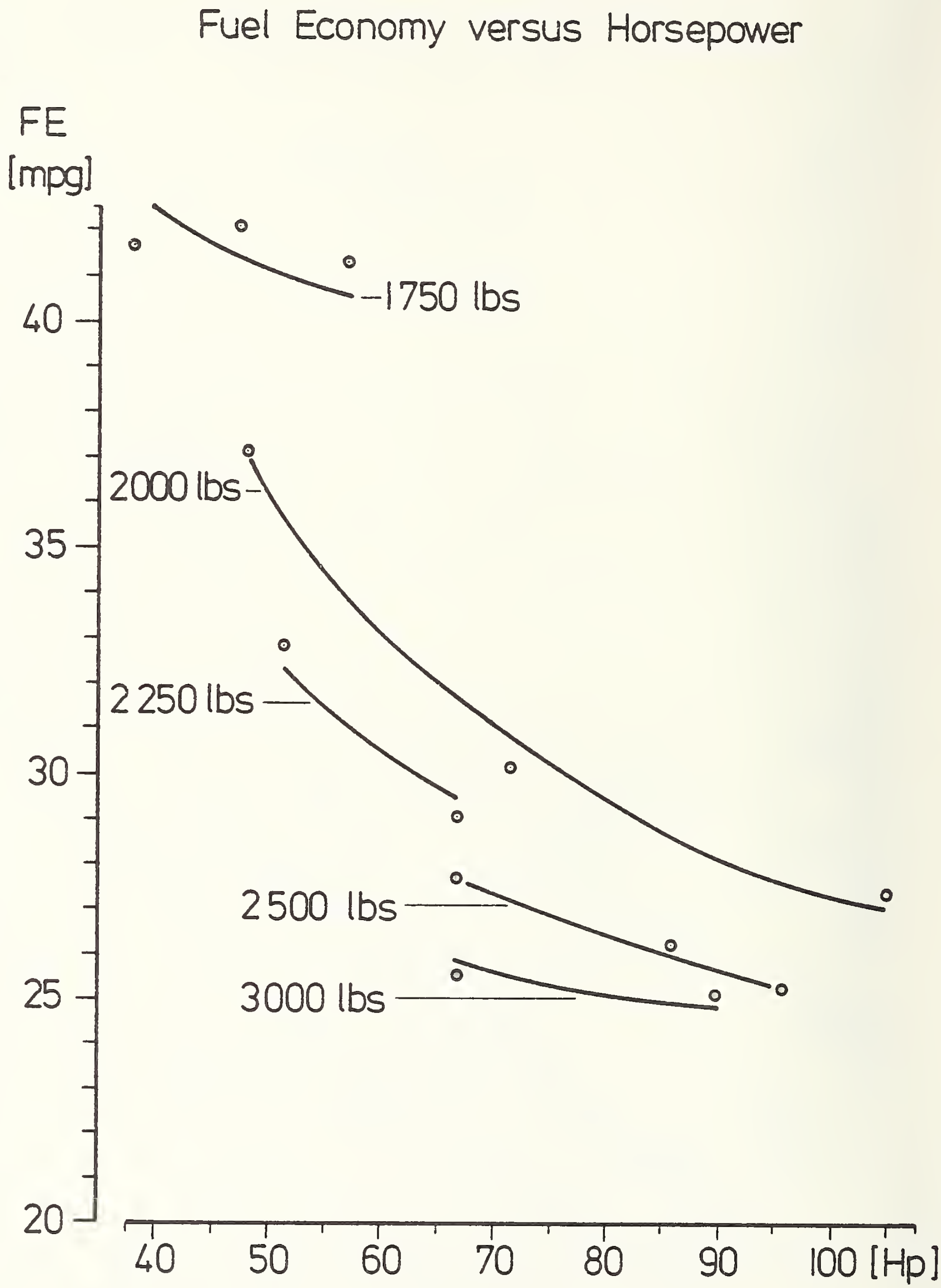
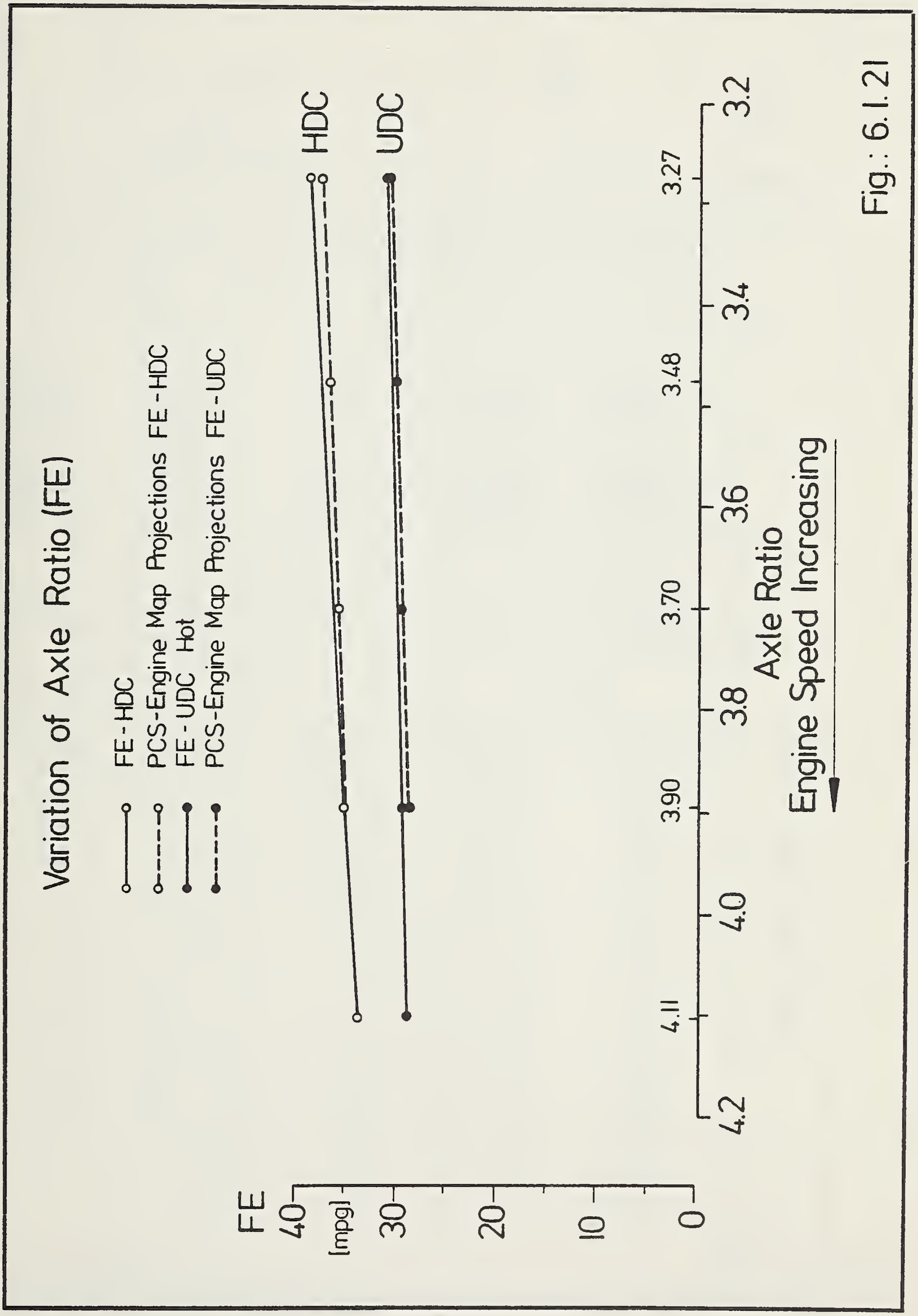
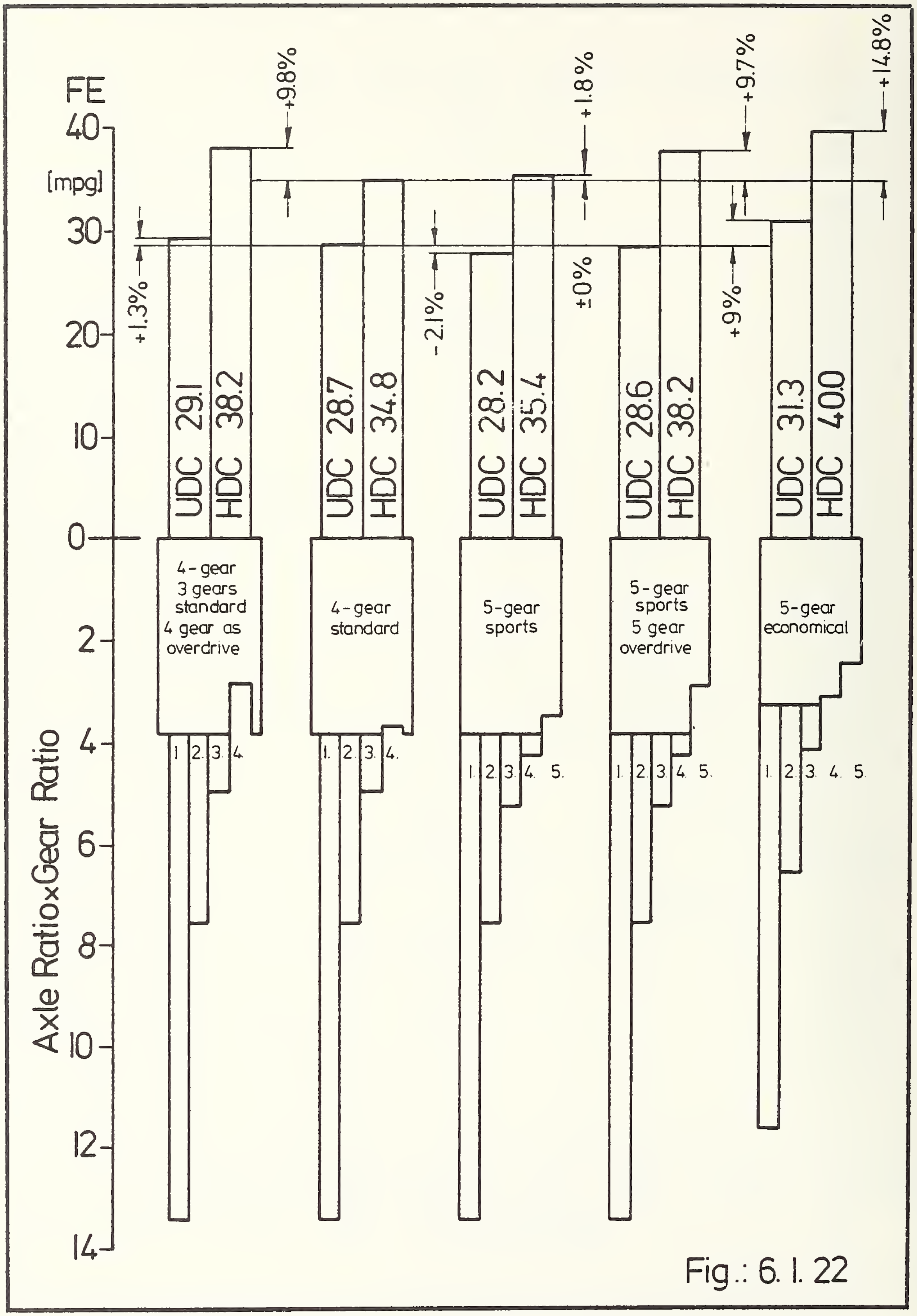

Fuel Economy versus Horsepower
FE
[mpg]
40
35
30
25
20
1750 lbs
2000 lbs
2 250 lbs
2 500 lbs
3000 lbs
40
50
60
70
80
90
100 [Hp]

Fig.: 6. 1. 20



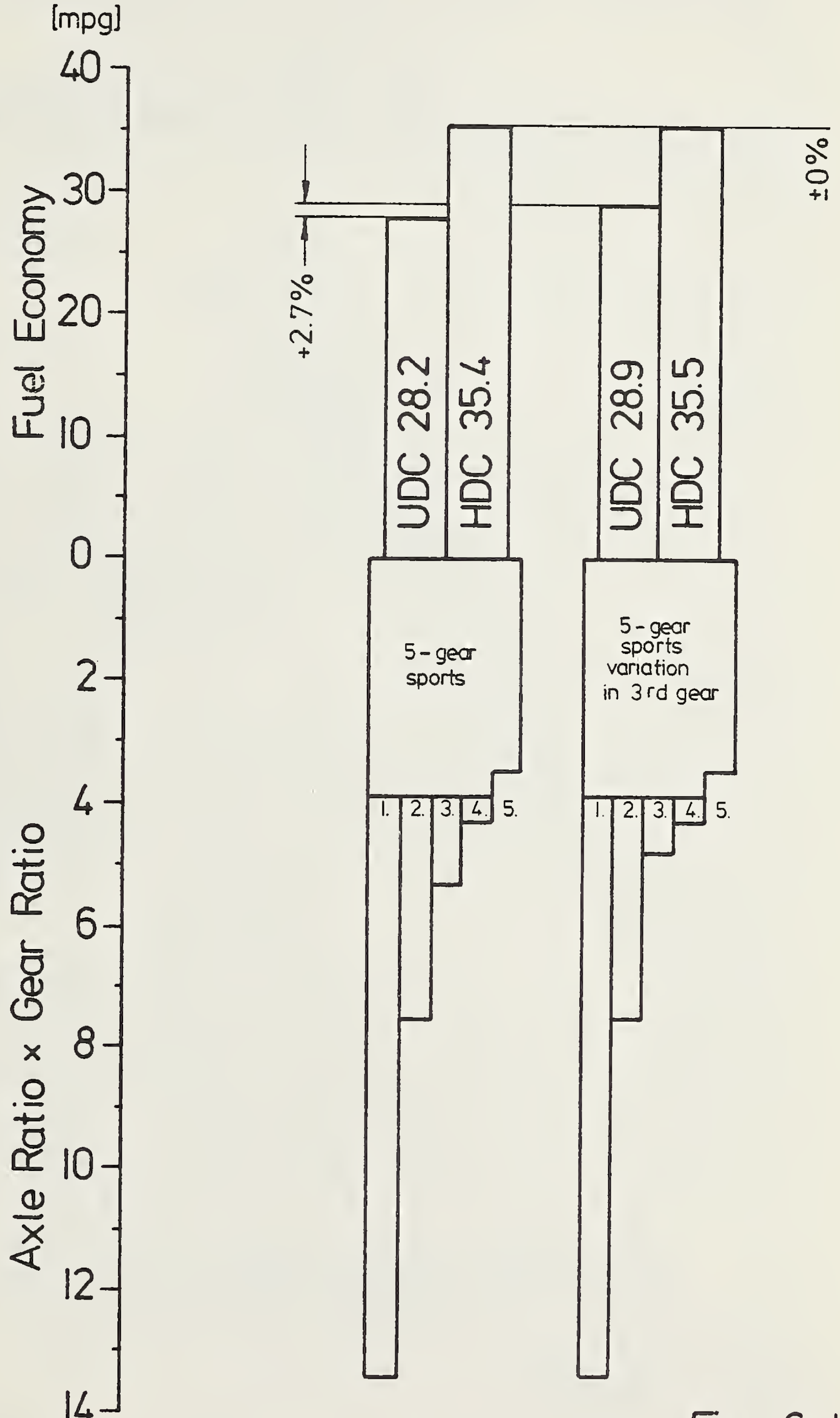
Variation of Axle Ratio (FE)
FE - HDC
PCS-Engine Map Projections FE - HDC
FE - UDC Hot
PCS-Engine Map Projections FE - UDC
HDC
UDC
FE [mpg]
40
30
20
10
0
4.2
4.11
4.0
3.90
3.8
3.70
3.6
3.48
3.4
3.27
3.2
Axle Ratio
Engine Speed Increasing

Fig.: 6.1.21



FE
[mpg]
40
30
20
10
0
2
4
6
8
10
12
14
Axle Ratio×Gear Ratio
+1.3%
+9.8%
-2.1%
±0%
+1.8%
+9%
+9.7%
+14.8%
UDC 29.1
HDC 38.2
UDC 28.7
HDC 34.8
UDC 28.2
HDC 35.4
UDC 28.6
HDC 38.2
UDC 31.3
HDC 40.0
4-gear 3 gears standard 4 gear as overdrive
4-gear standard
5-gear sports
5-gear sports 5 gear overdrive
5-gear economical
1 2. 3. 4.
1. 2. 3. 4.
1. 2. 3. 4. 5.
1. 2. 3. 4. 5.
1. 2. 3. 4. 5.

Fig.: 6. 1. 22



[mpg]
Fuel Economy
40
30
20
10
0
2
4
6
8
10
12
14
Axle Ratio x Gear Ratio
+2.7%
±0%
UDC 28.2
HDC 35.4
UDC 28.9
HDC 35.5
5 - gear sports
5 - gear sports variation in 3rd gear
1.
2.
3.
4.
5.

Fig.: 6. I. 23

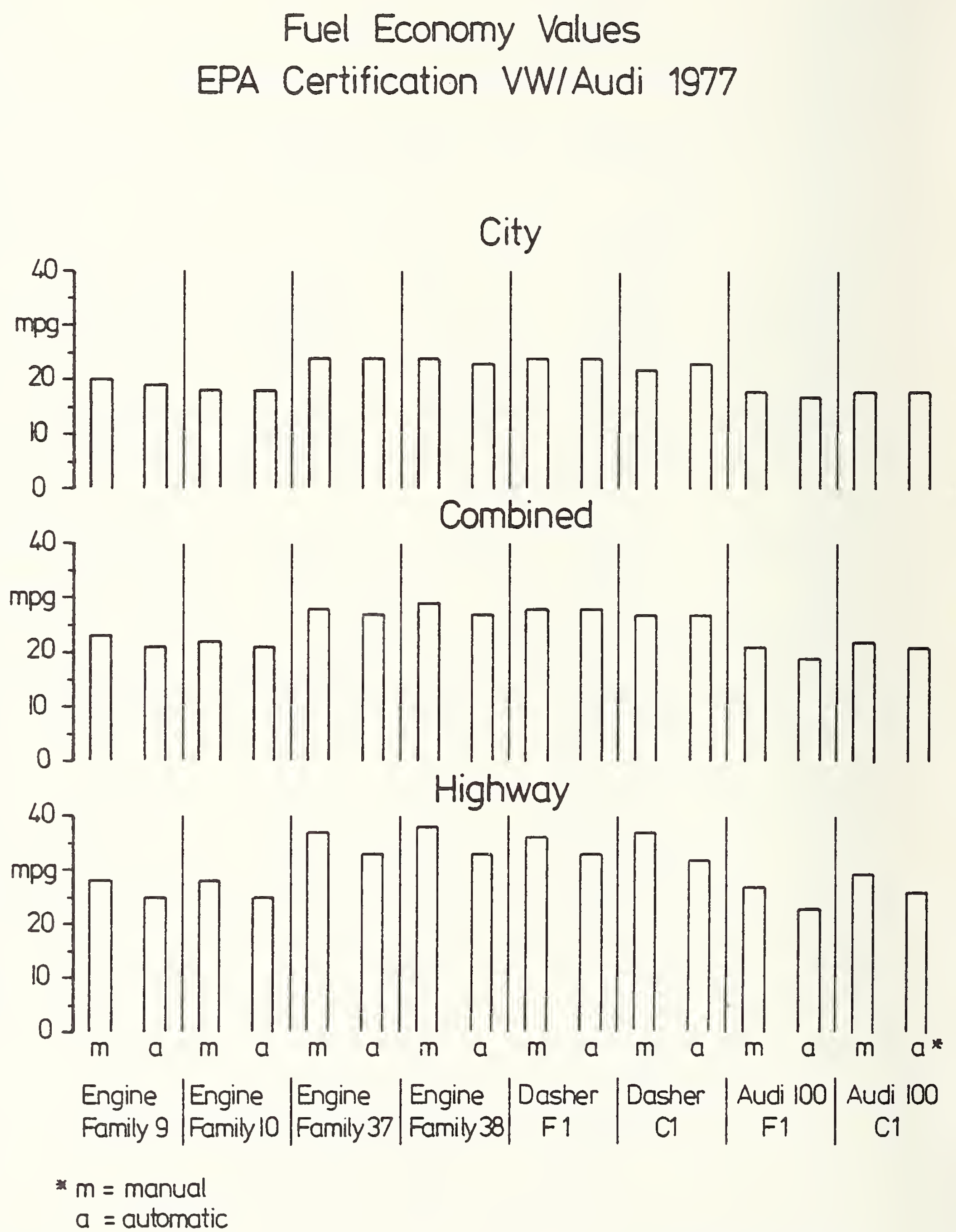
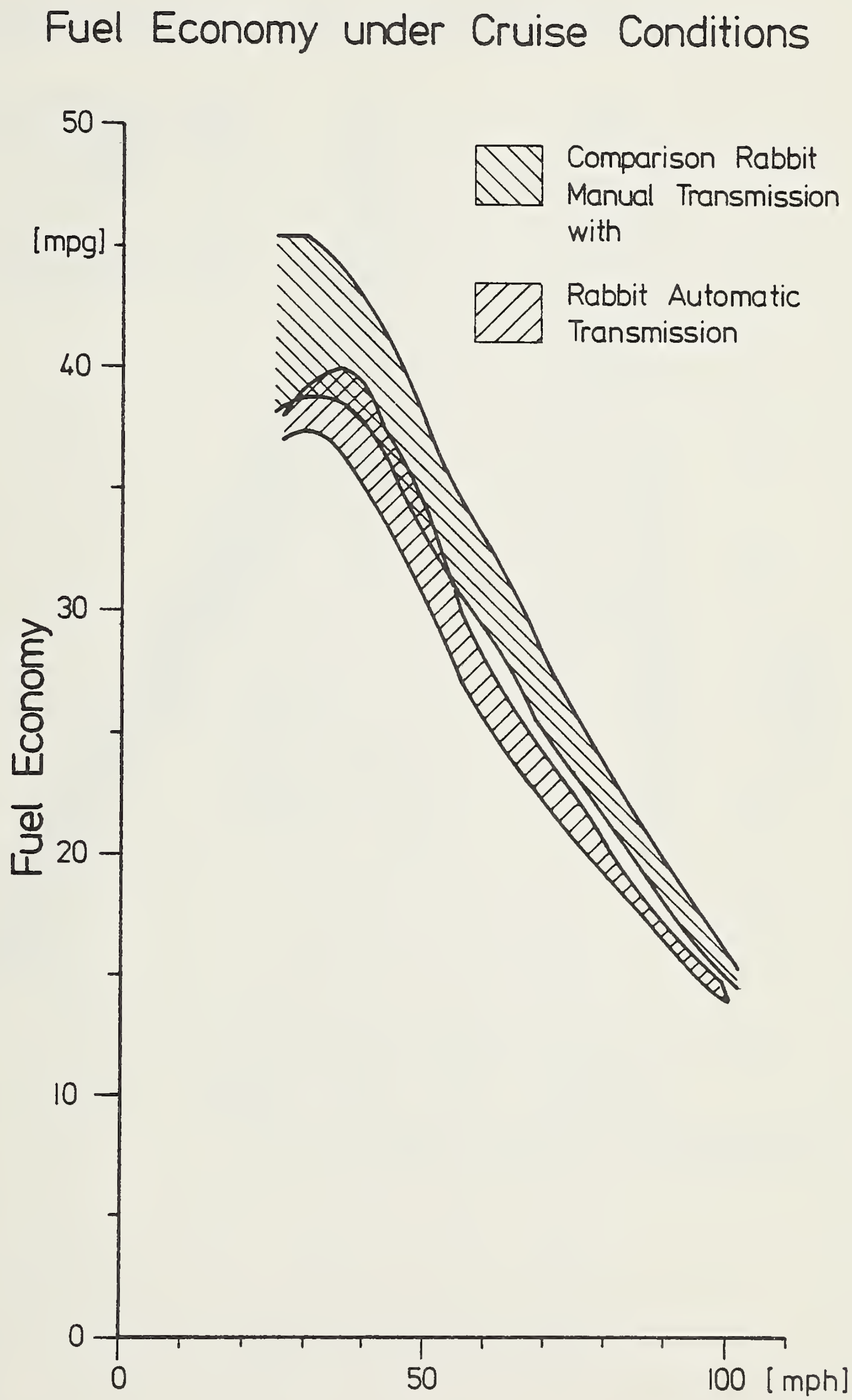

Fuel Economy Values
EPA Certification VW/Audi 1977
City
Combined
Highway
40
mpg
20
10
0
m a m a m a m a m a m a m a m a*
Engine Family 9
Engine Family 10
Engine Family 37
Engine Family 38
Dasher F 1
Dasher C1
Audi 100 F1
Audi 100 C1
* m = manual
a = automatic

Fig.: 6. 1. 24



Fuel Economy under Cruise Conditions
Comparison Rabbit Manual Transmission with
Rabbit Automatic Transmission
Fuel Economy
[mpg]
50
40
30
20
10
0
0
50
100 [mph]

Fig.: 6. 1. 25

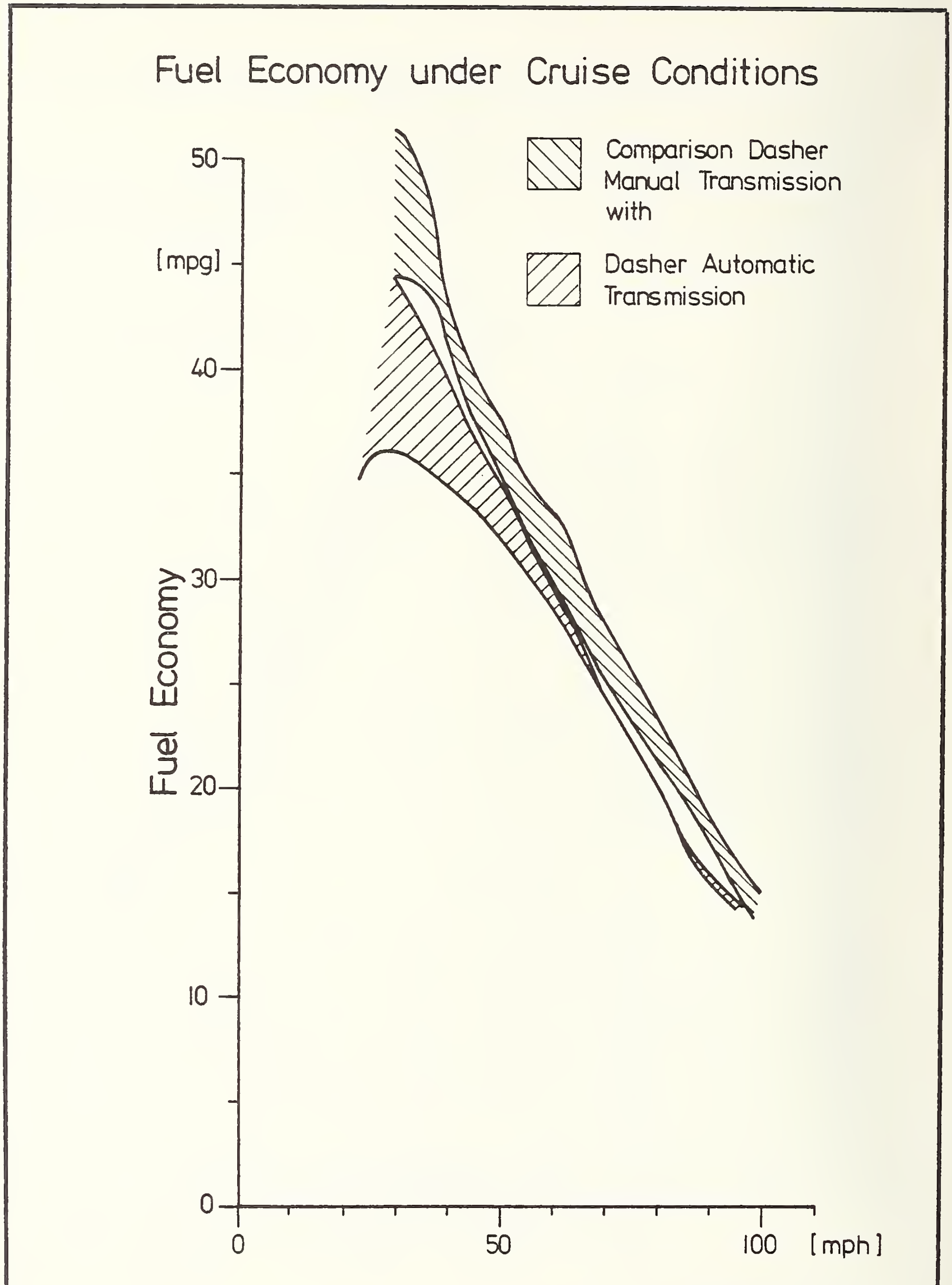

Fuel Economy under Cruise Conditions
Comparison Dasher Manual Transmission with
Dasher Automatic Transmission
50
[mpg]
40
30
20
10
0
Fuel Economy
0
50
100
[mph]

Fig.: 6. I. 26

Road Load Horsepower Demand as a Function of Speed

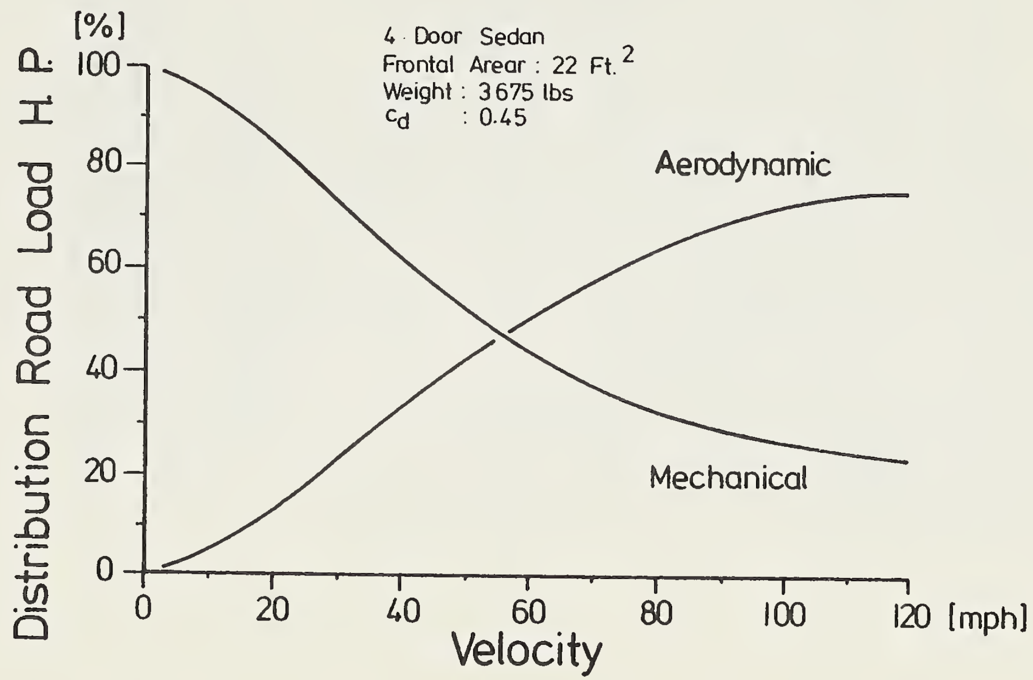
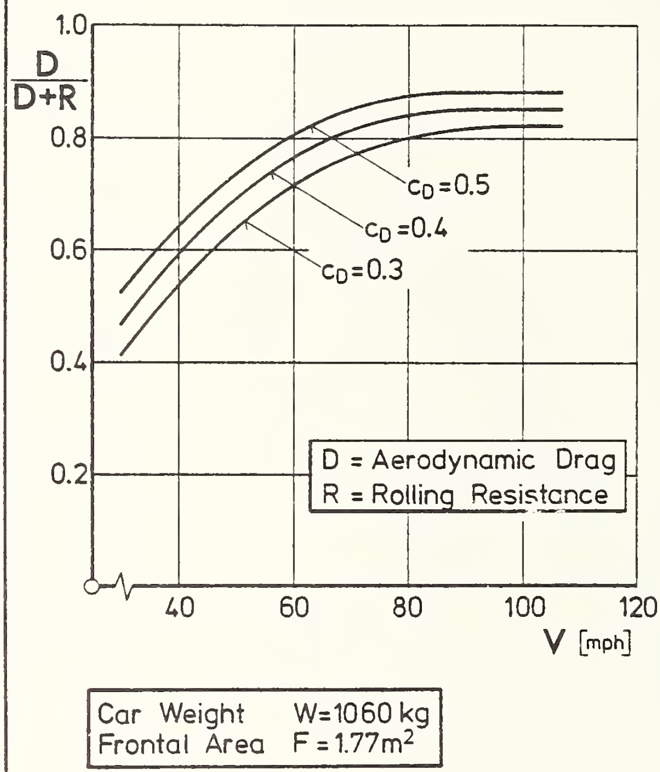
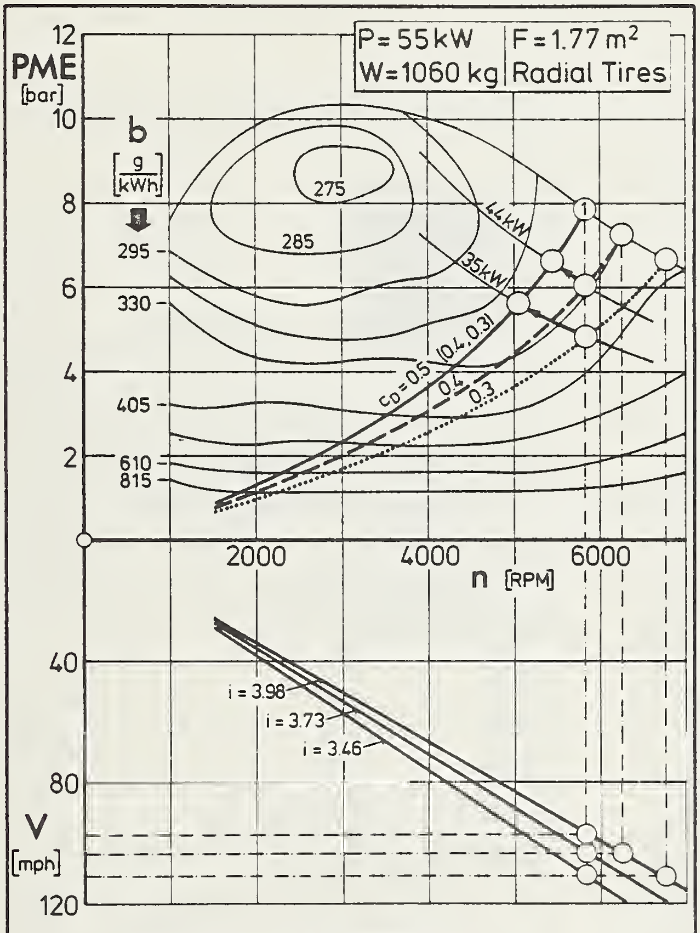


Fig.: 6.1.27 [2]



Aerodynamic Drag Related to Total External Drag

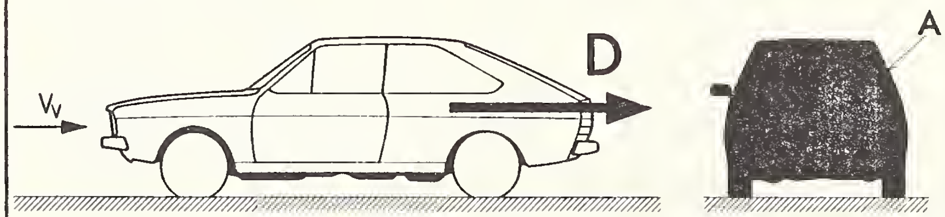
Fig.: 6. I. 28 [3]



Road Load Curve and SFC-Map

FE 75 21 03

Fig.: 6. 1. 29 [3]



$$D = c_D \cdot A \cdot \frac{\varrho}{2} V_V^2$$

$$\boxed{c_D = \frac{D}{A \cdot \frac{\varrho}{2} V_V^2}}$$

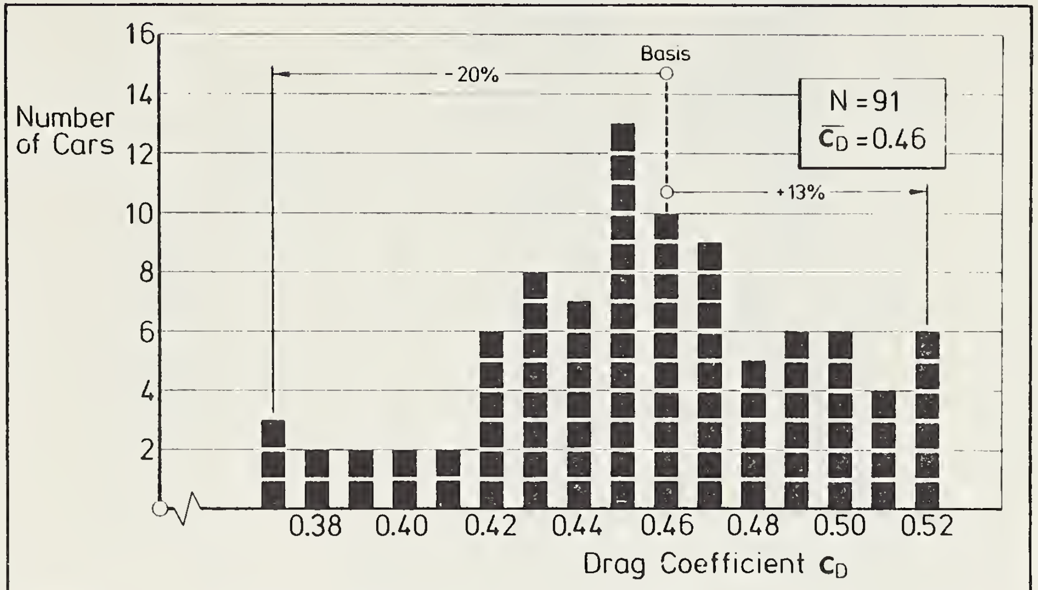
$$P_D = D \cdot V_V = c_D \cdot A \cdot \frac{\varrho}{2} V_V^3$$

D = Aerodynamic Drag
c_D = Aerodynamic Drag Coefficient
A = Projected Frontal Area
ϱ = Air Density
V_V = Vehicle Velocity
P_D = Aerodynamic Drag Force

Definition of
Aerodynamic Drag Coefficient c_D

FE
76 18 14

Fig.: 6. I. 30



Drag Coefficients of N=91
European Passenger Cars (1968 to 1976)

FE
76 18 27

Fig.: 6. I. 31

History of Progress in Drag Reduction

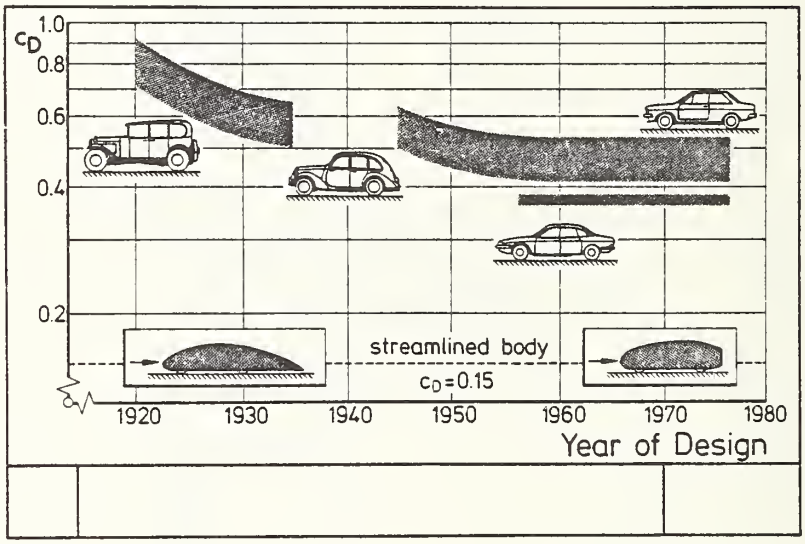
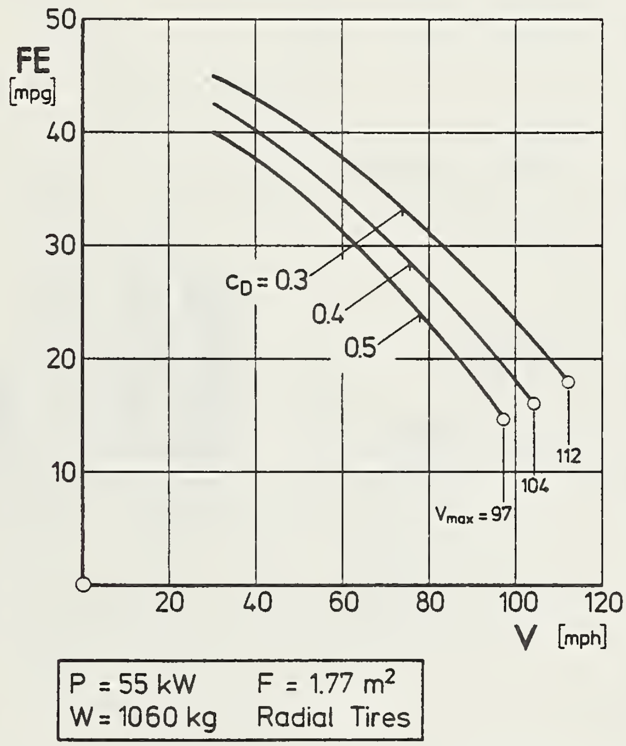
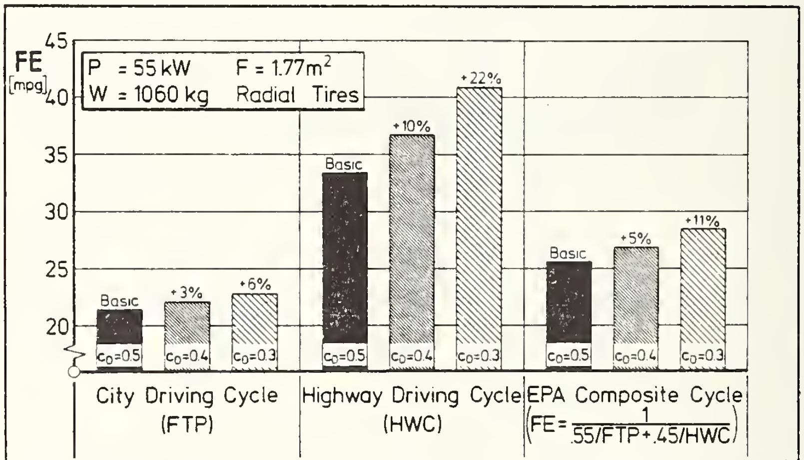


Fig.: 6.1.32 [3]



Steady State
Fuel Economy

Fig.: 6.1.33 [3]



Fuel Economy in Standard Cycles

Fig.: 6.1.34 [3]

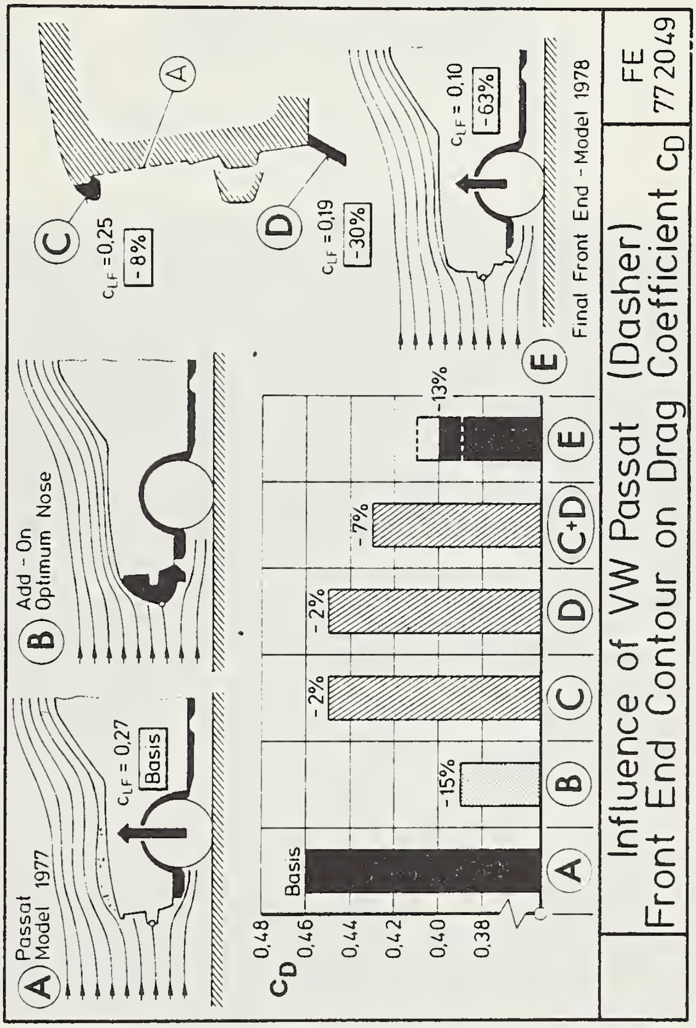
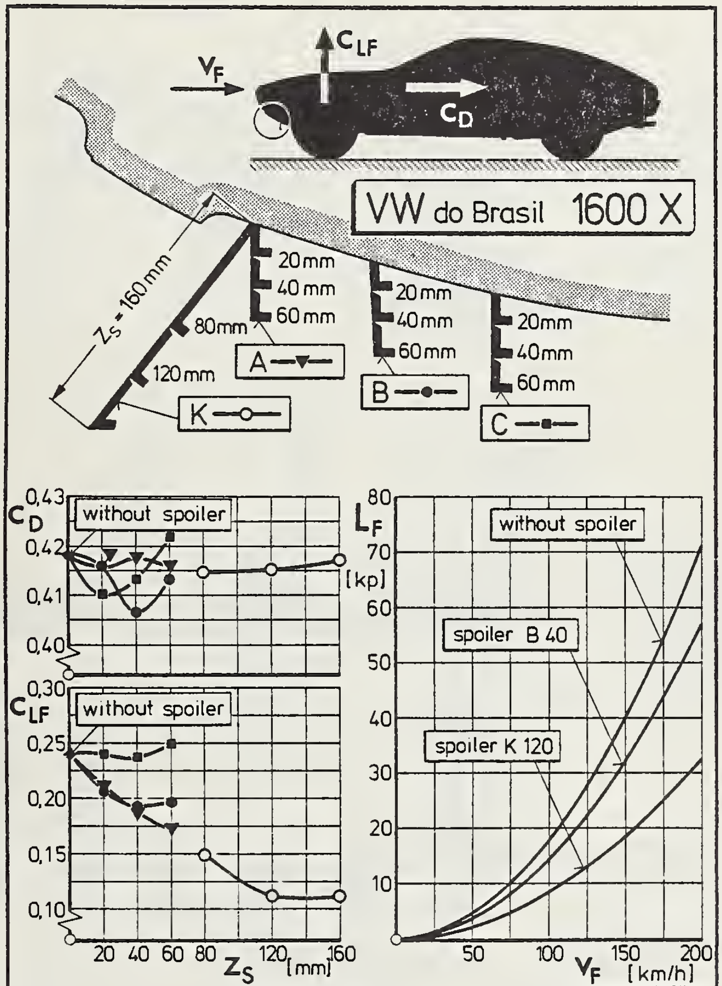

A Passat Model 1977
B Add - On Optimum Nose
c_{LF} = 0,27
Basis
C
c_{LF} = 0,25
-8%
A
D
c_{LF} = 0,19
-30%
c_{LF} = 0,10
-63%
E
Final Front End - Model 1978
c_D
0,48
0,46
0,44
0,42
0,40
0,38
Basis
-15%
-2%
-2%
-7%
-13%
A
B
C
D
C+D
E
Influence of VW Passat (Dasher)
Front End Contour on Drag Coefficient c_D
FE
772049

Fig.: 6. I. 35

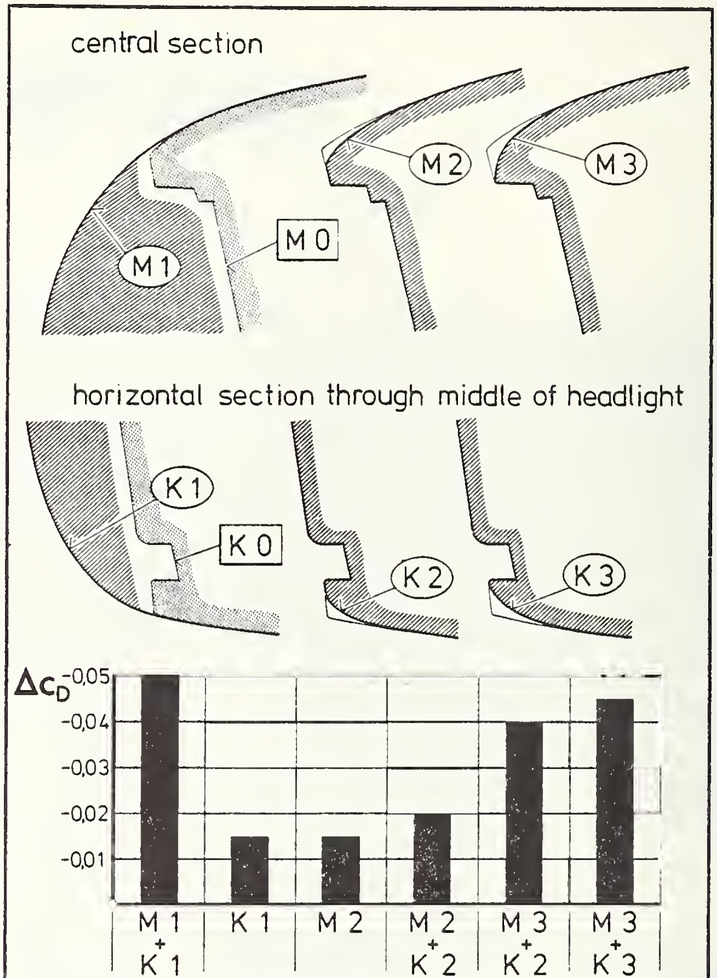
Streamlines passing the VW Dasher in the Volks-wagen full scale wind tunnel

Fig.: 6.1.36 [3]



	Influence of Front Spoiler Design	FE 72 25 15

Fig.: 6.1.37 [3]



	Influence of Different Front Shapes	FE 72 25 14

Fig.: 6.1.38 [3]

Influence of Front Edge Shape on Static Pressure and Drag Coefficient

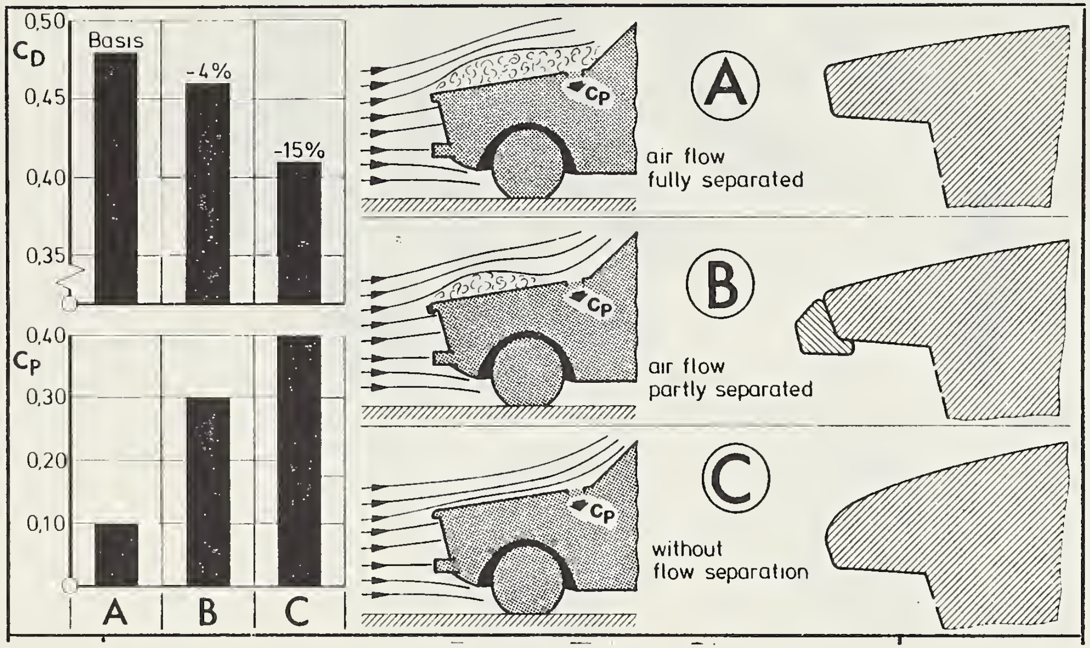
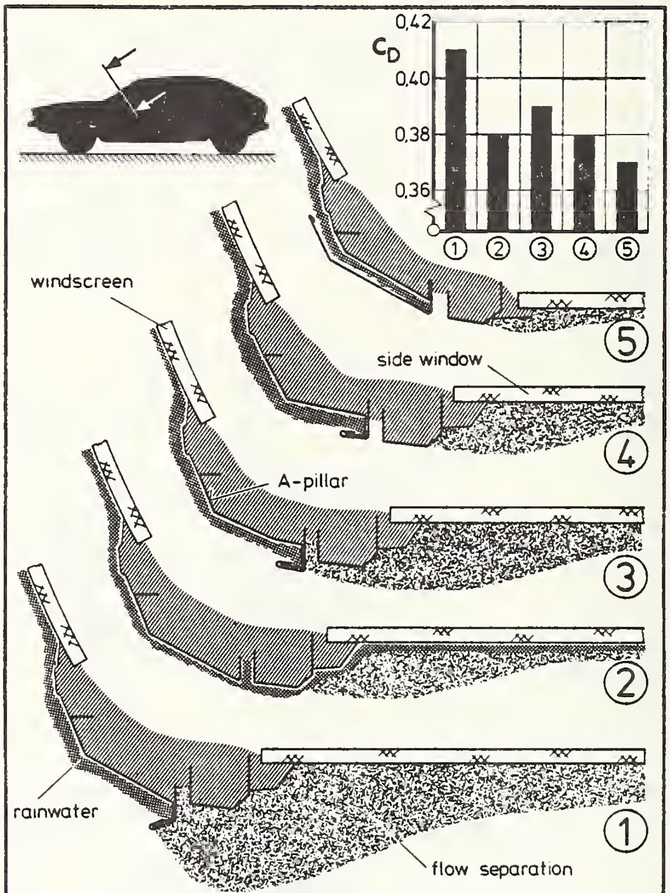
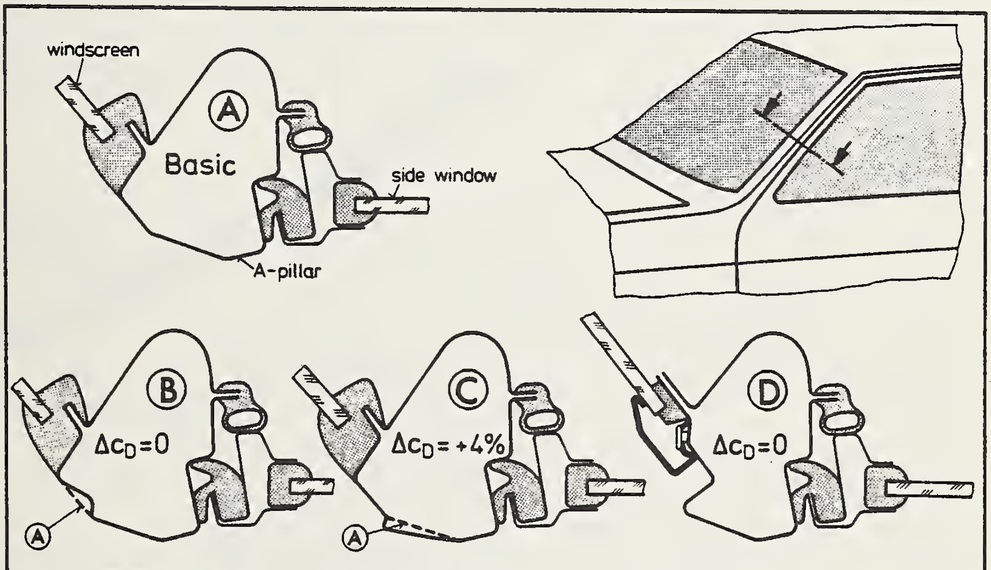


Fig.: 6. I. 39 [3]



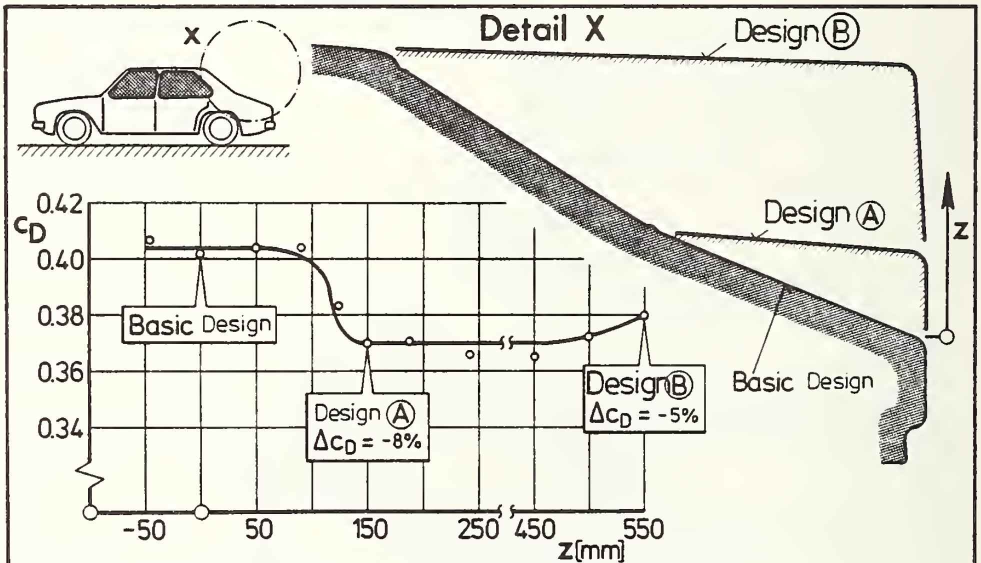
	Influence of A-Pillar Design	FE 72 25 16

Fig.: 6.1.40 [3]



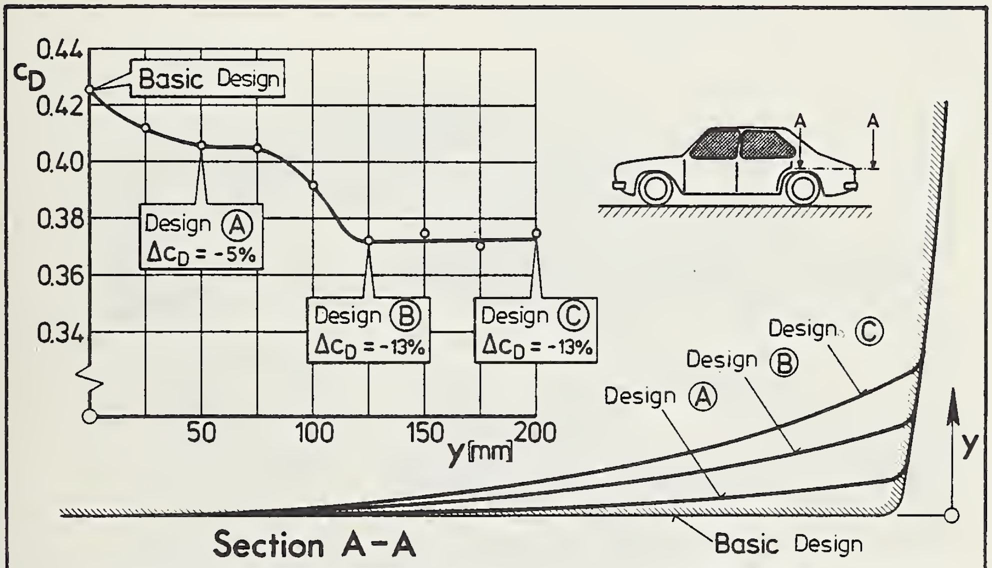
Influence of A-Pillar Shape on Drag Coefficient c_D

Fig.: 6.1.41 [3]



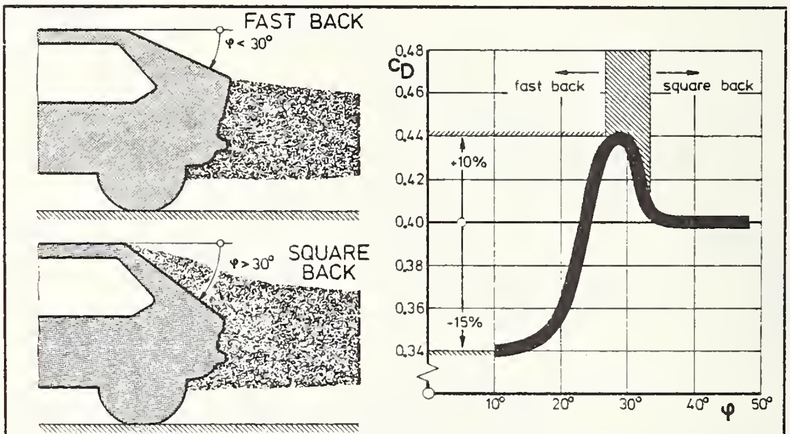
Influence of Rear End Elevation z on Drag Coefficient c_D

Fig.: 6. I. 42 [3]



Influence of Side Panel Contraction y on Drag Coefficient c_D

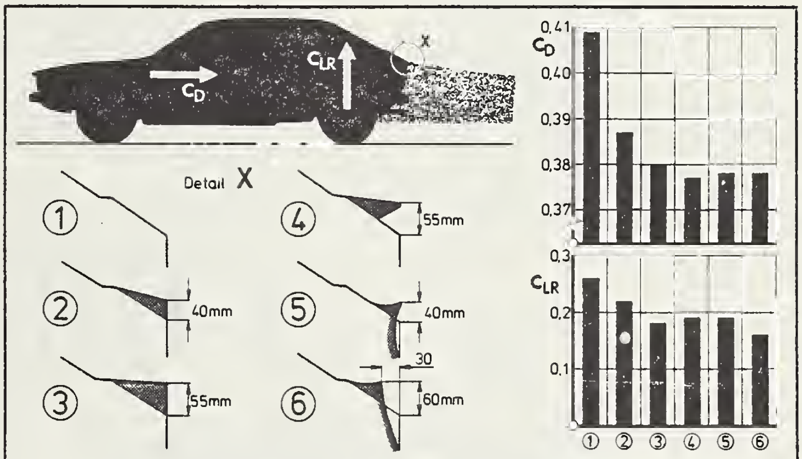
Fig.: 6.1.43 [3]



Influence of Rear End Inclination Angle φ on Drag Coefficient c_D

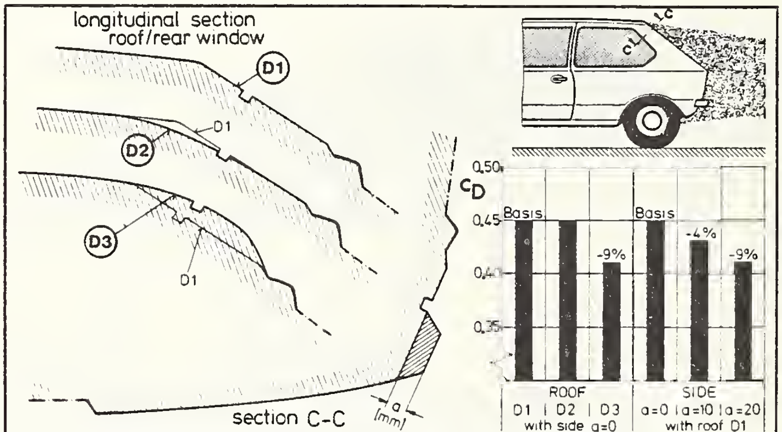
FE 752121

Fig.: 6 I. 44 [3]



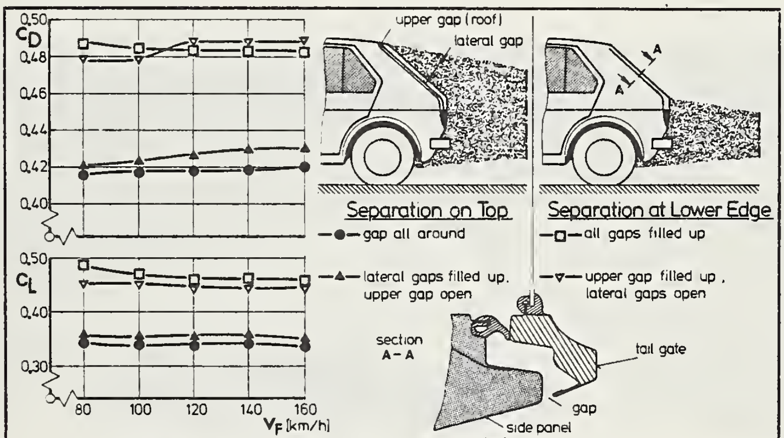
	Influence of Rear Spoiler Design	FE 72 25 18

Fig.: 6.1.45[3]



	Influence of Roof / Rear Window Junction and Rear Pillar Shape on Coefficient c_D	FE 75 2122

Fig.: 6.1.46 [3]



	Influence of Tail Gate-to-Body Gap on Drag Coefficient c_D and Lift Coefficient c_L	FE 75 21 23

Fig.: 6. 1. 47 [3]

Streamlines passing the VW Scirocco in the Volks-
wagen full scale wind tunnel

Fig.: 6. 1. 48 [3]

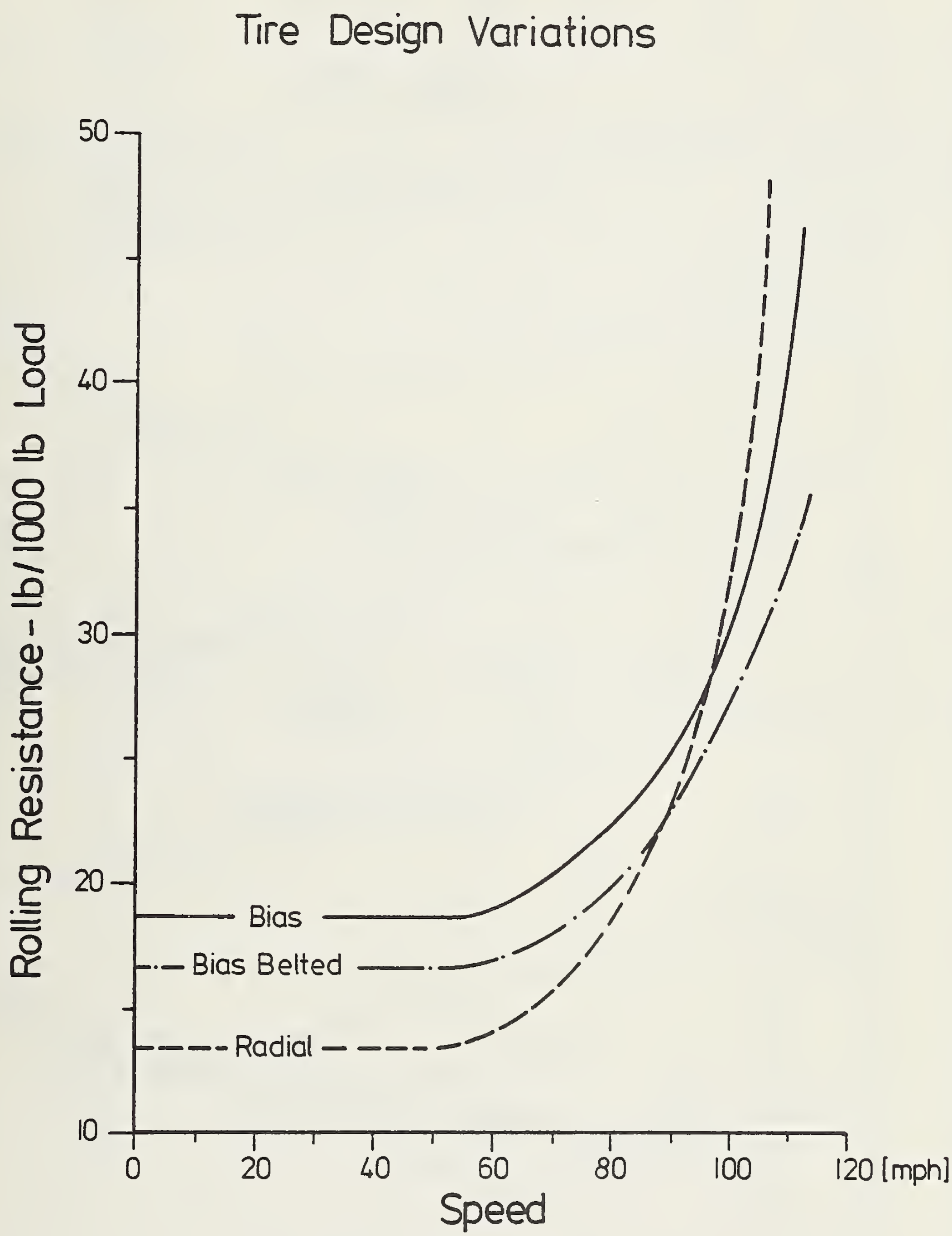

Tire Design Variations
50
40
30
20
10
Rolling Resistance - lb/1000 lb Load
Bias
Bias Belted
Radial
0
20
40
60
80
100
120 [mph]
Speed

Fig.: 6. 1. 49 [2]

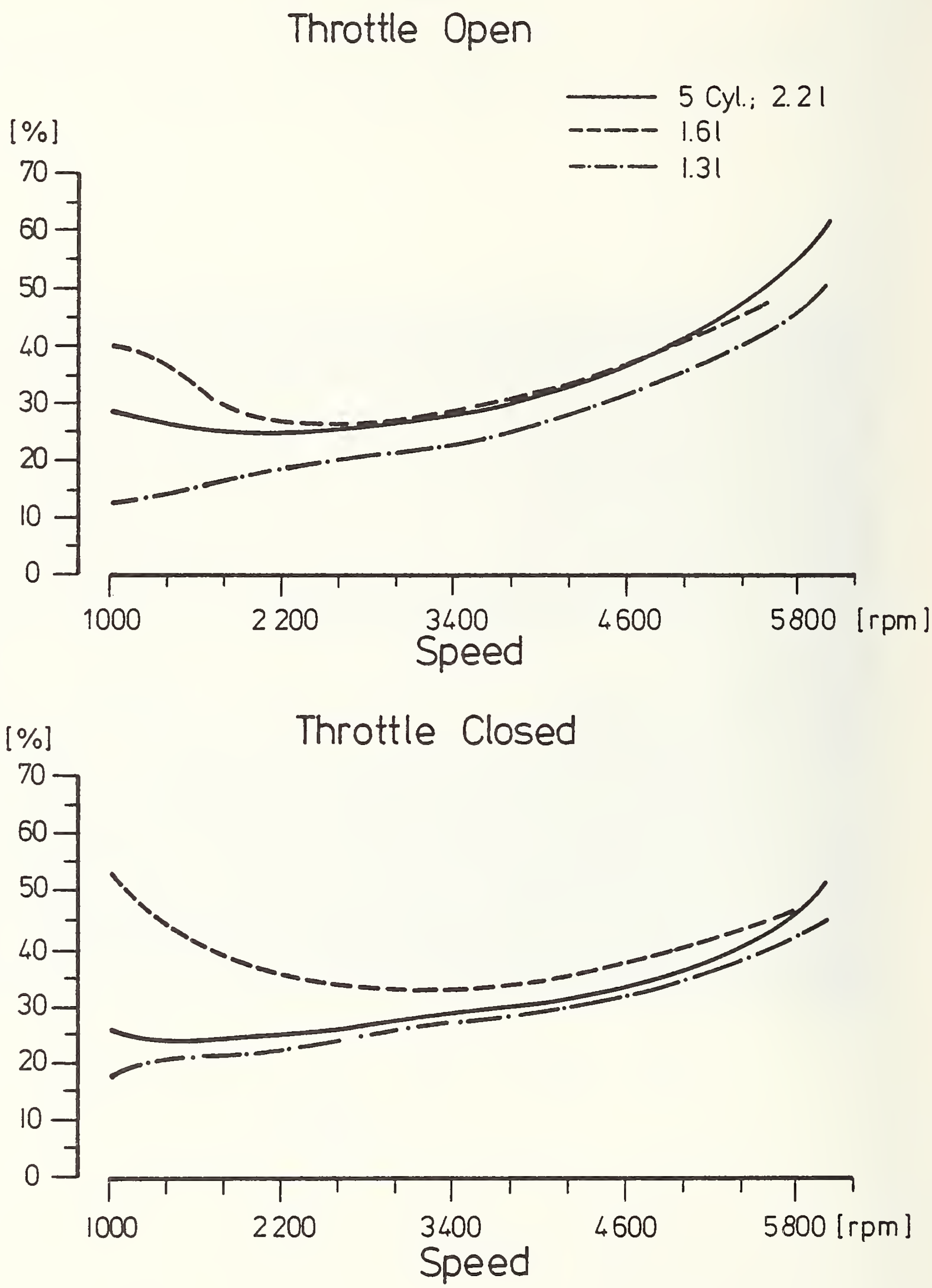

Throttle Open
5 Cyl.; 2.2 l
1.6 l
1.3 l
[%]
70
60
50
40
30
20
10
0
1000
2 200
3400
4600
5800 [rpm]
Speed
Throttle Closed
[%]
70
60
50
40
30
20
10
0
1000
2200
3400
4600
5800 [rpm]
Speed

Fig.: 6.1.50

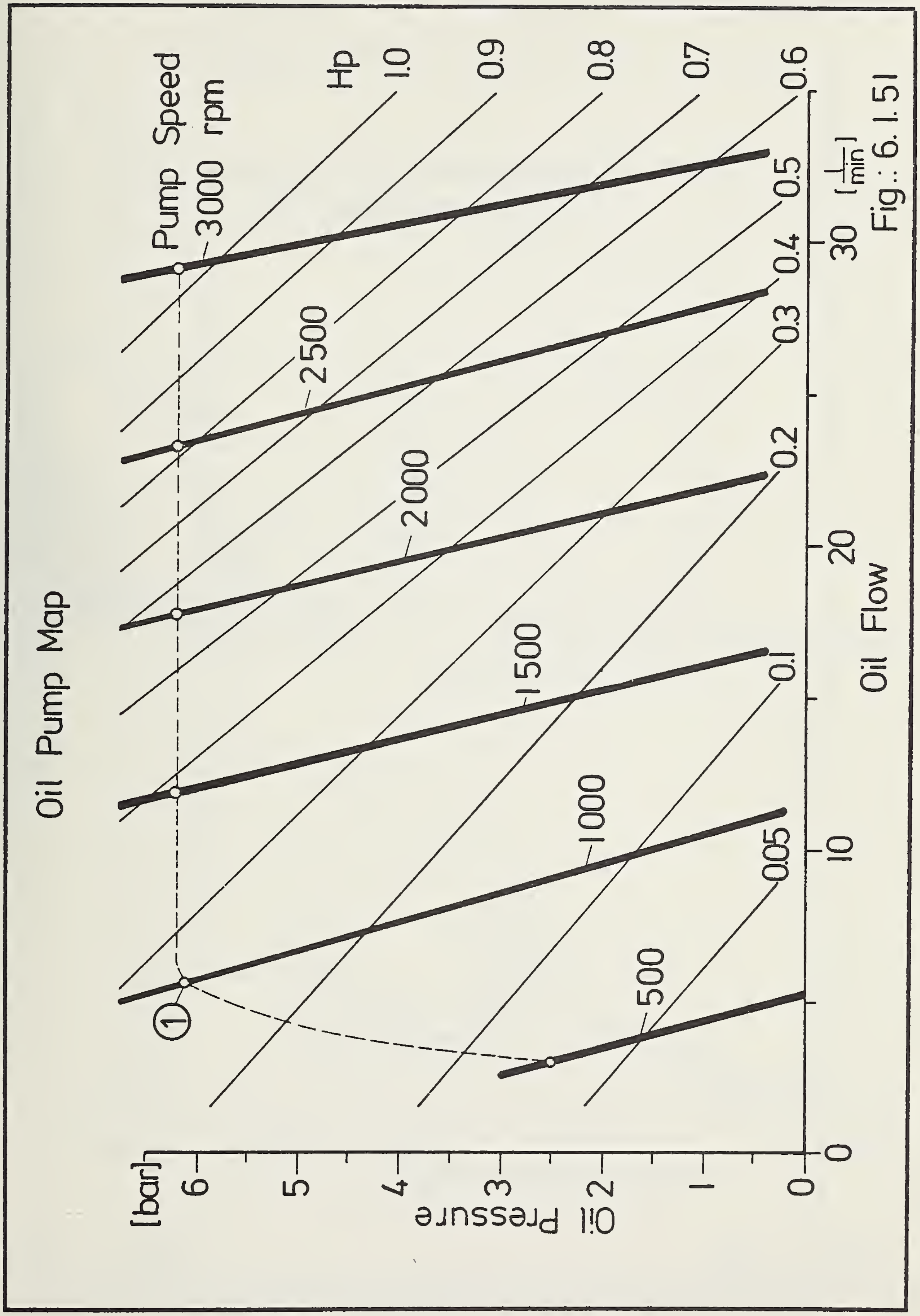

Oil Pump Map
Pump Speed
3000 rpm
2500
2000
1500
1000
500
1
Hp
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.05
[bar]
Oil Pressure
0
1
2
3
4
5
6
Oil Flow
0
10
20
30
[l/min]

Fig.: 6.1.51

Power Requirements for Accessories in Standard - Size Car

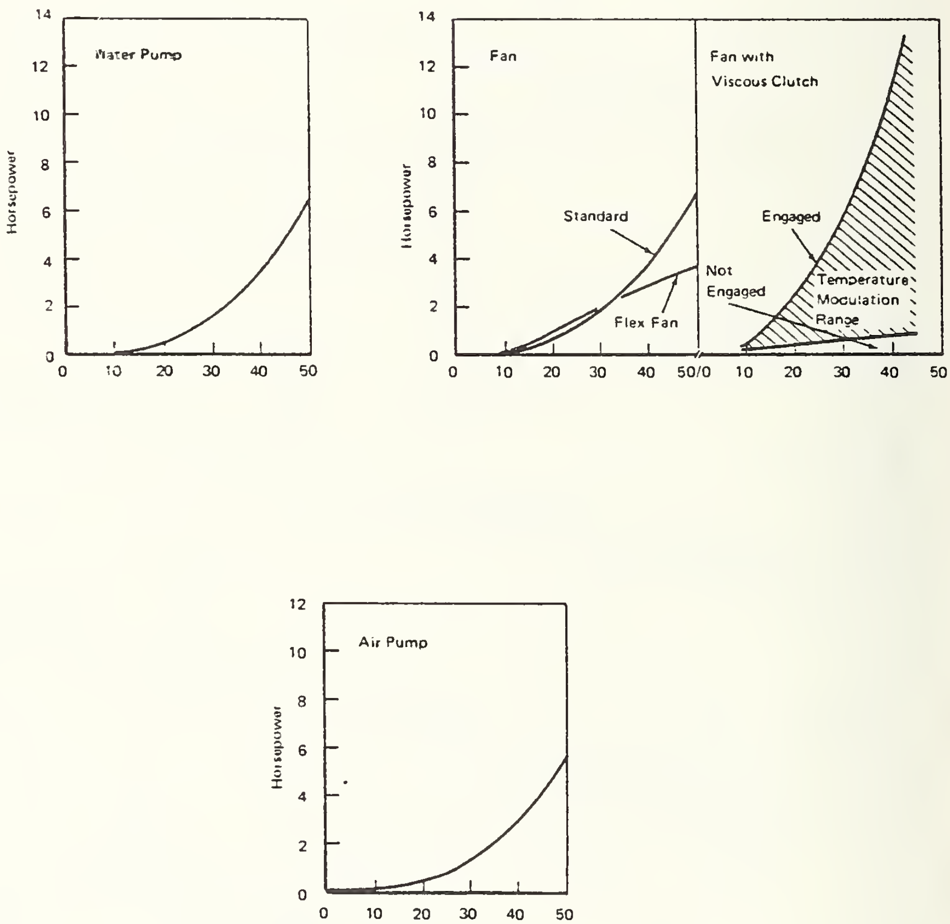


Fig.: 6. I. 52 [2]

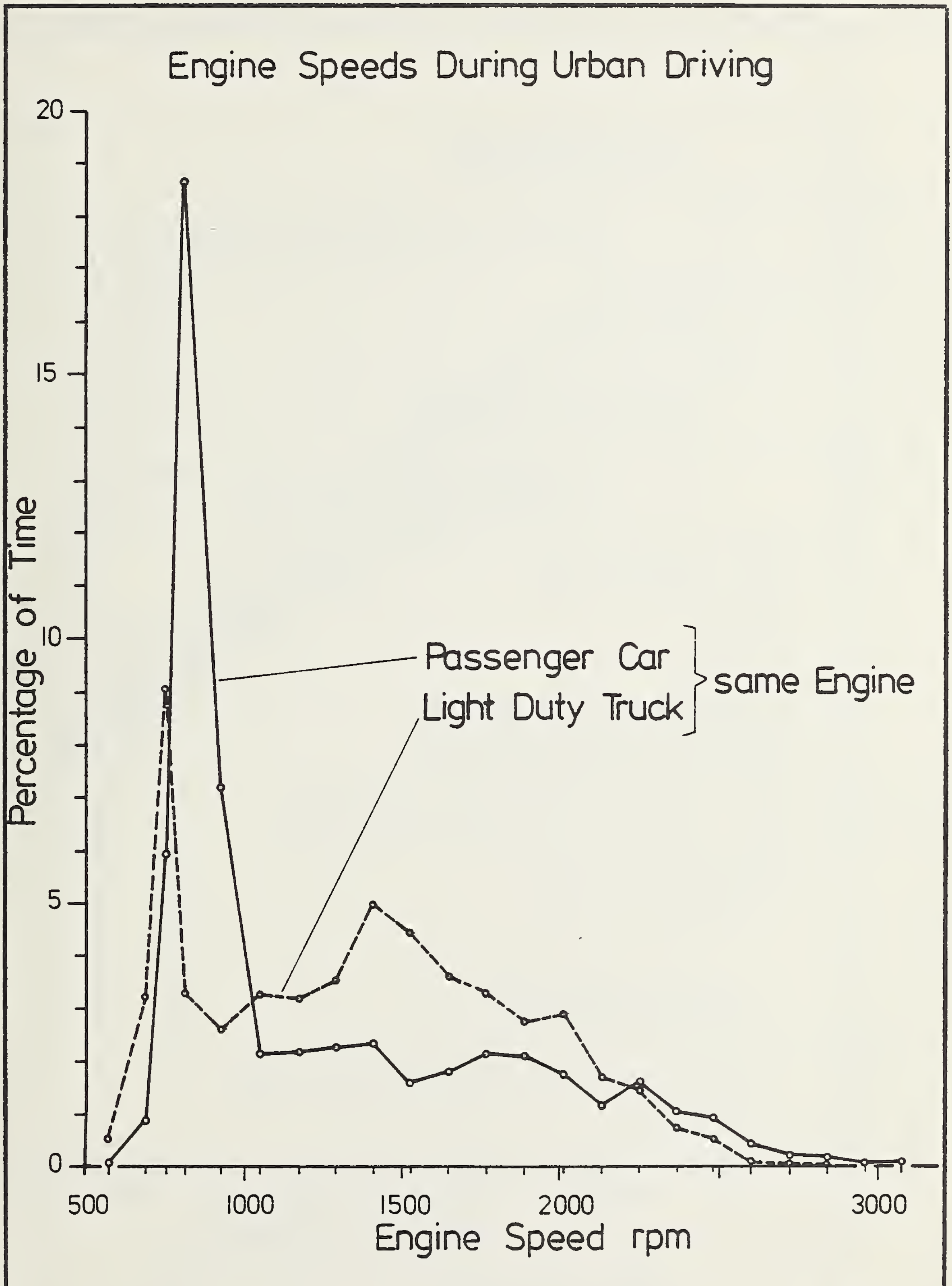

Engine Speeds During Urban Driving
20
15
10
5
0
Percentage of Time
Passenger Car
Light Duty Truck
same Engine
500
1000
1500
2000
3000
Engine Speed rpm

Fig.: 6. I. 53

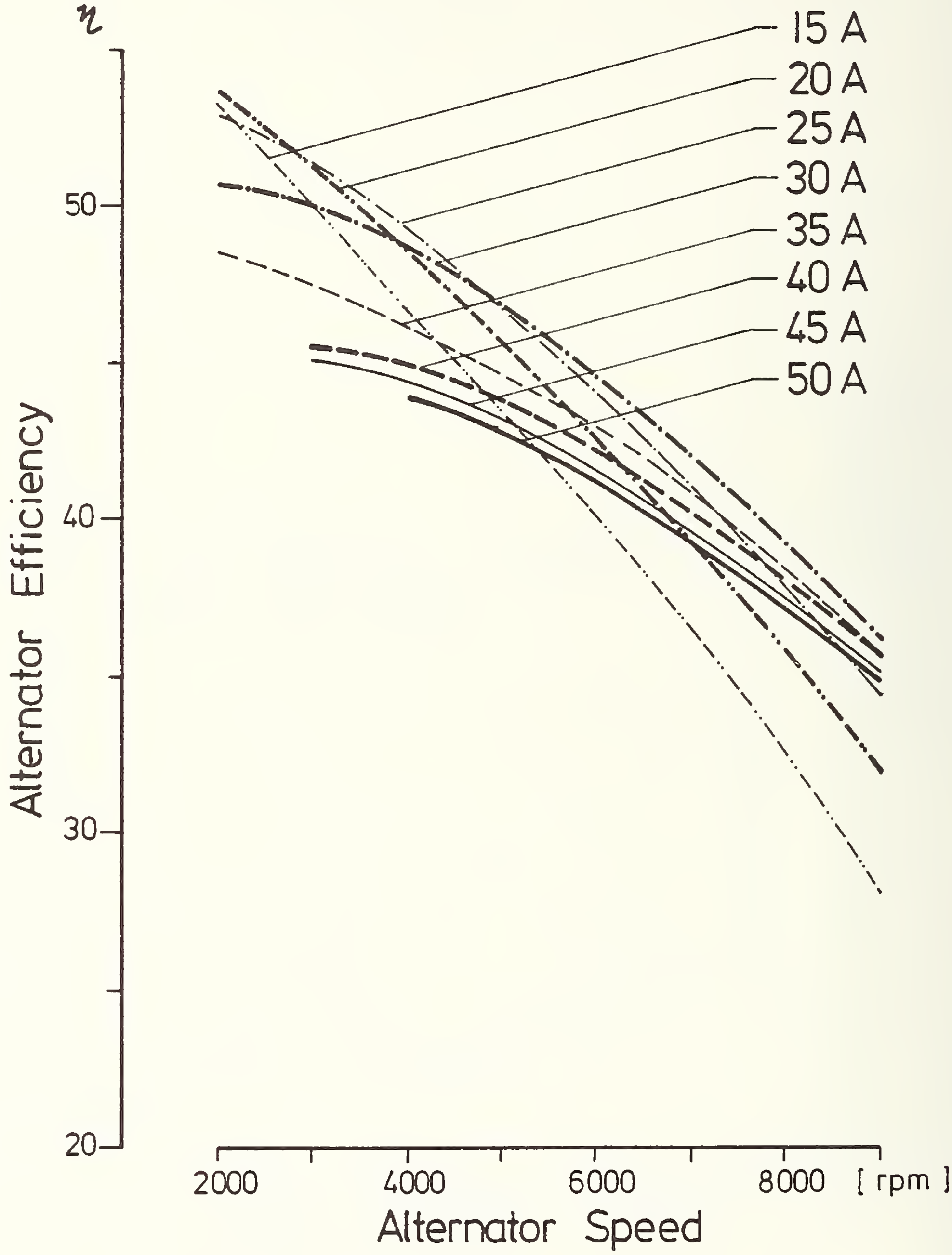

η
50
40
30
20
Alternator Efficiency
15 A
20 A
25 A
30 A
35 A
40 A
45 A
50 A
2000
4000
6000
8000
[rpm]
Alternator Speed

Fig.: 6. 1. 54

Fuel Economy with and without Air Conditioner

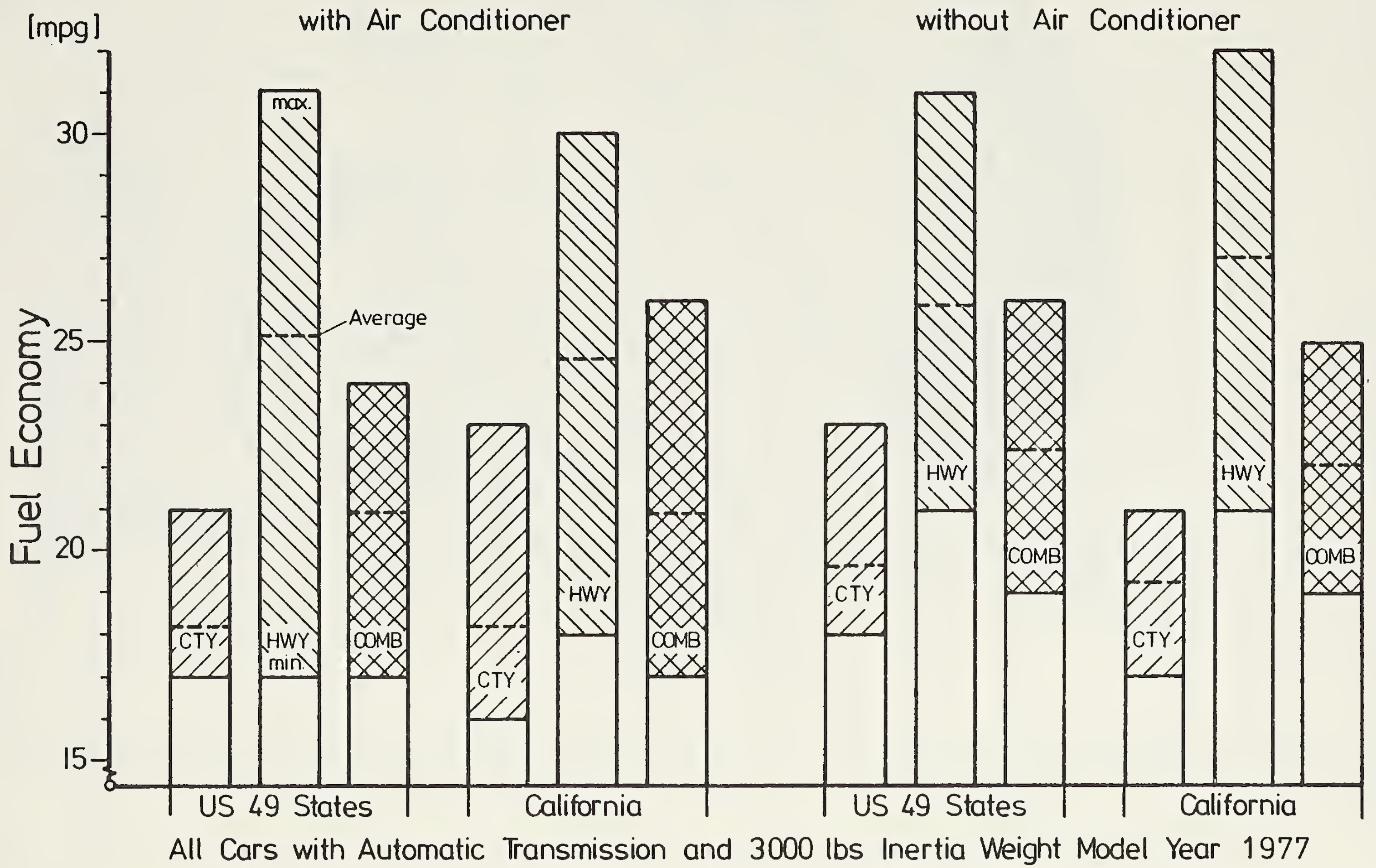
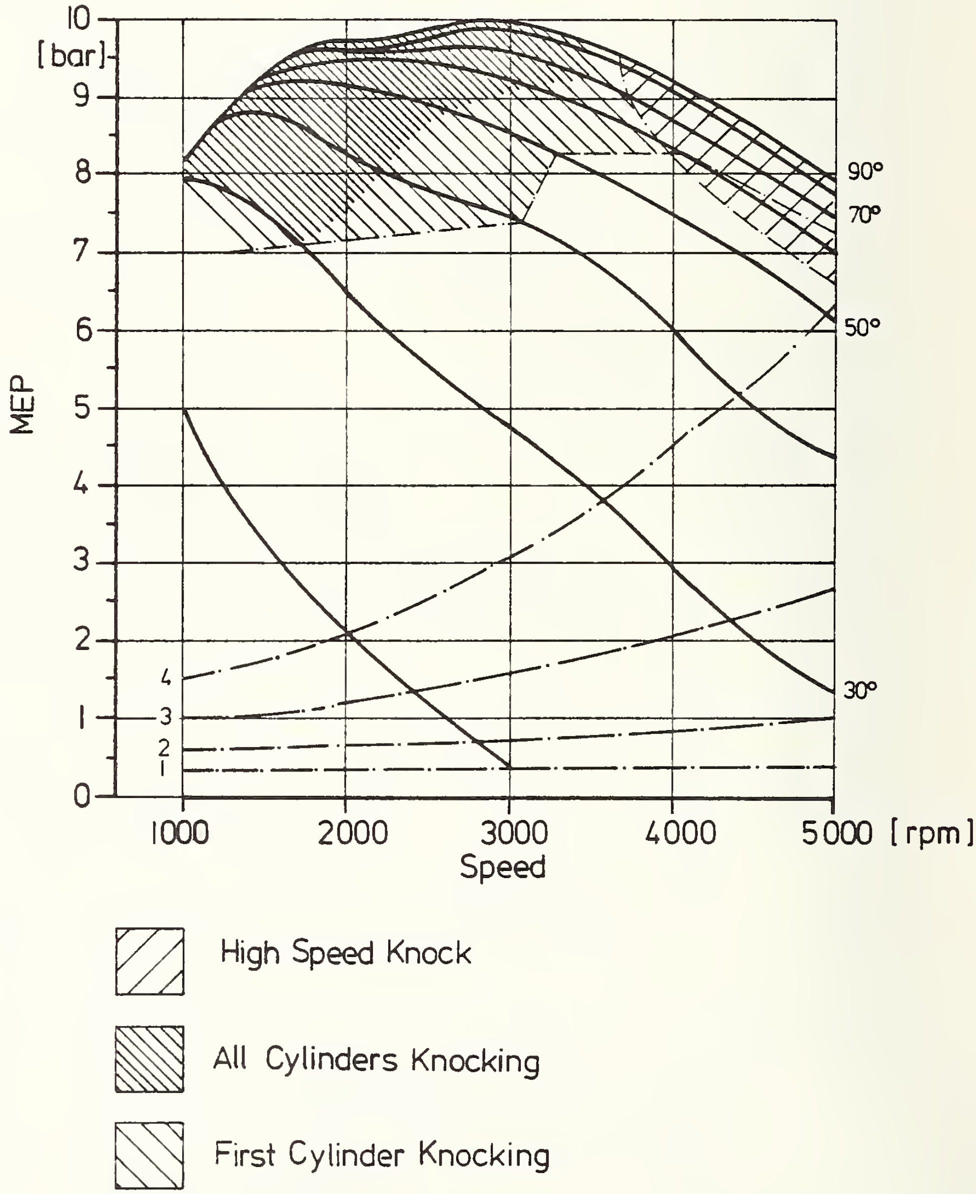


Fig.: 6.1.55



10
[bar]
9
8
7
6
5
4
3
2
1
0
MEP
4
3
2
1
1000
2000
3000
4000
5000 [rpm]
Speed
90°
70°
50°
30°
High Speed Knock
All Cylinders Knocking
First Cylinder Knocking

Fig.: 6. 1. 56

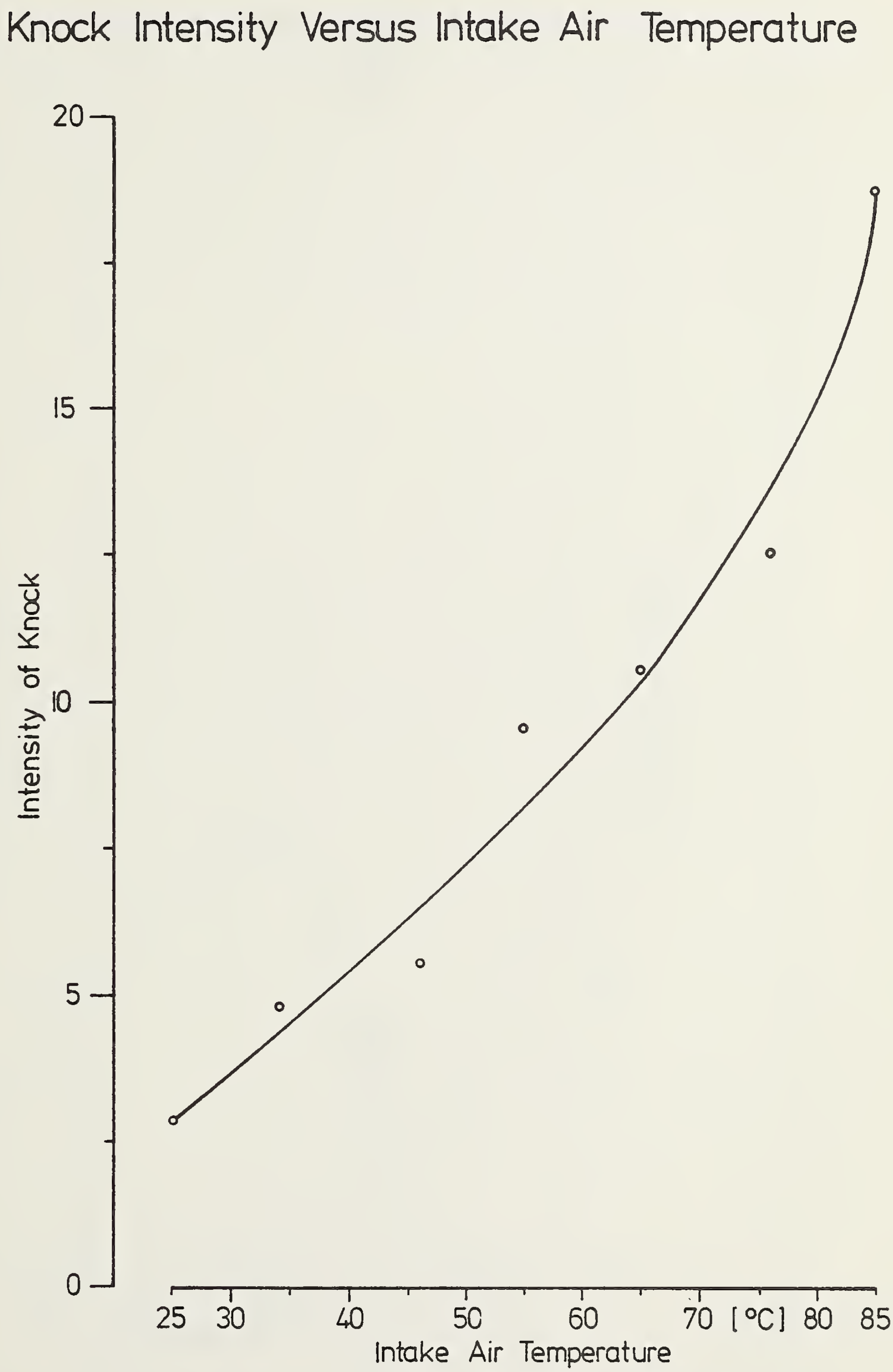

Knock Intensity Versus Intake Air Temperature
20
15
10
5
0
Intensity of Knock
25 30 40 50 60 70 [°C] 80 85
Intake Air Temperature

Fig.: 6. 1. 57[4]

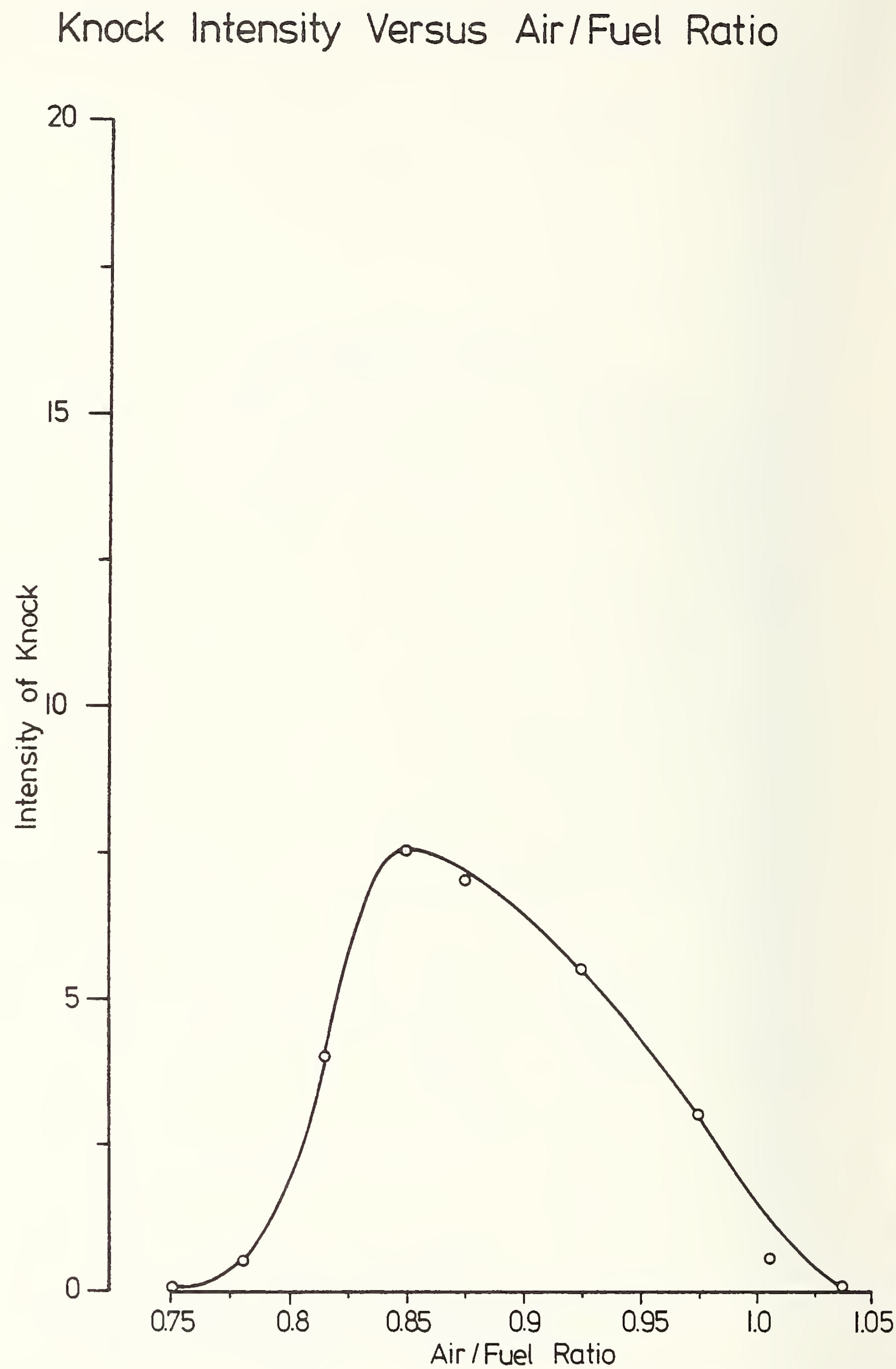

Knock Intensity Versus Air/Fuel Ratio
20
15
10
5
0
Intensity of Knock
0.75
0.8
0.85
0.9
0.95
1.0
1.05
Air / Fuel Ratio

Fig.: 6.1.58 [4]

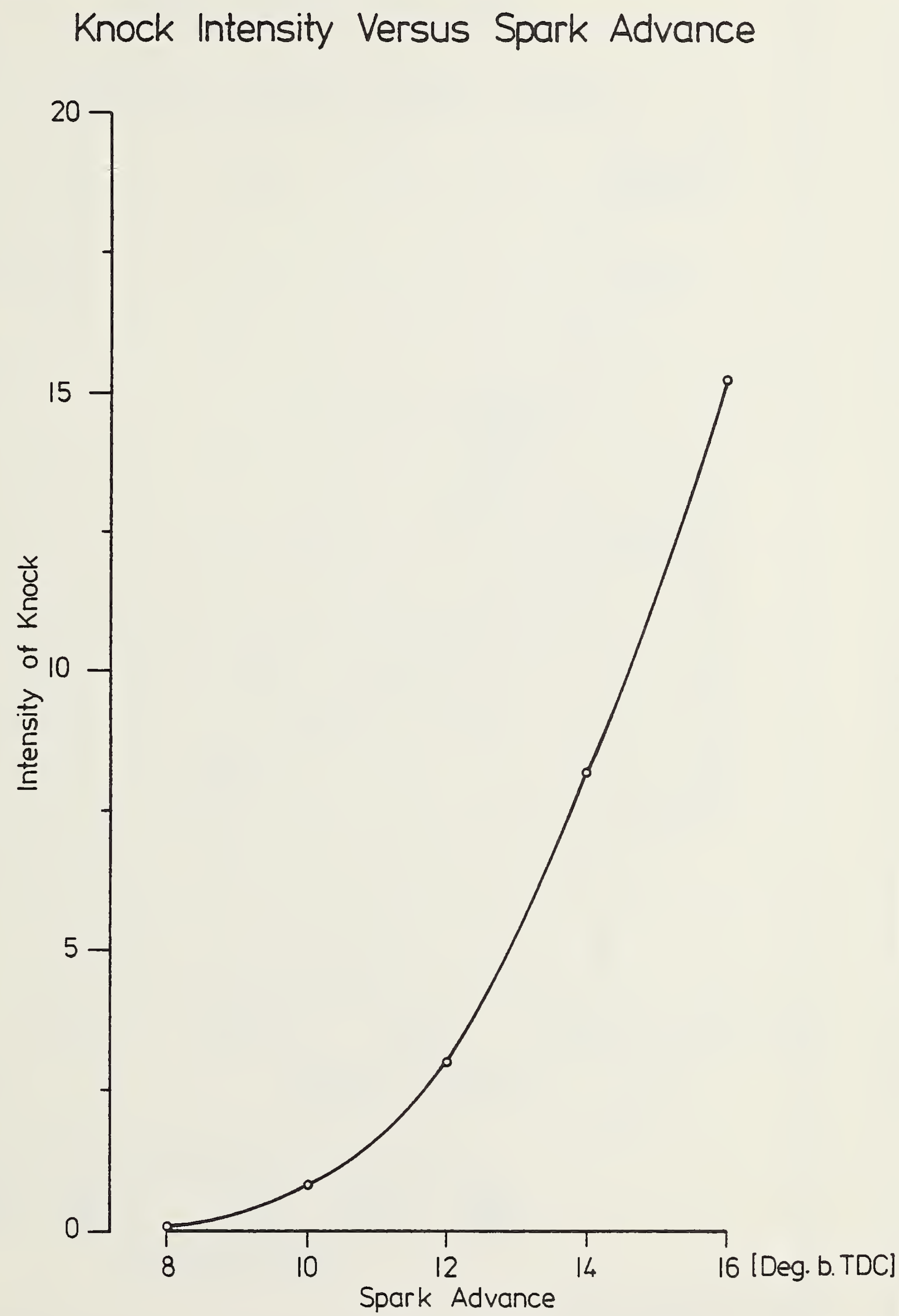

Knock Intensity Versus Spark Advance
Intensity of Knock
20
15
10
5
0
8
10
12
14
16
[Deg. b. TDC]
Spark Advance

Fig.: 6. 1. 59[4]

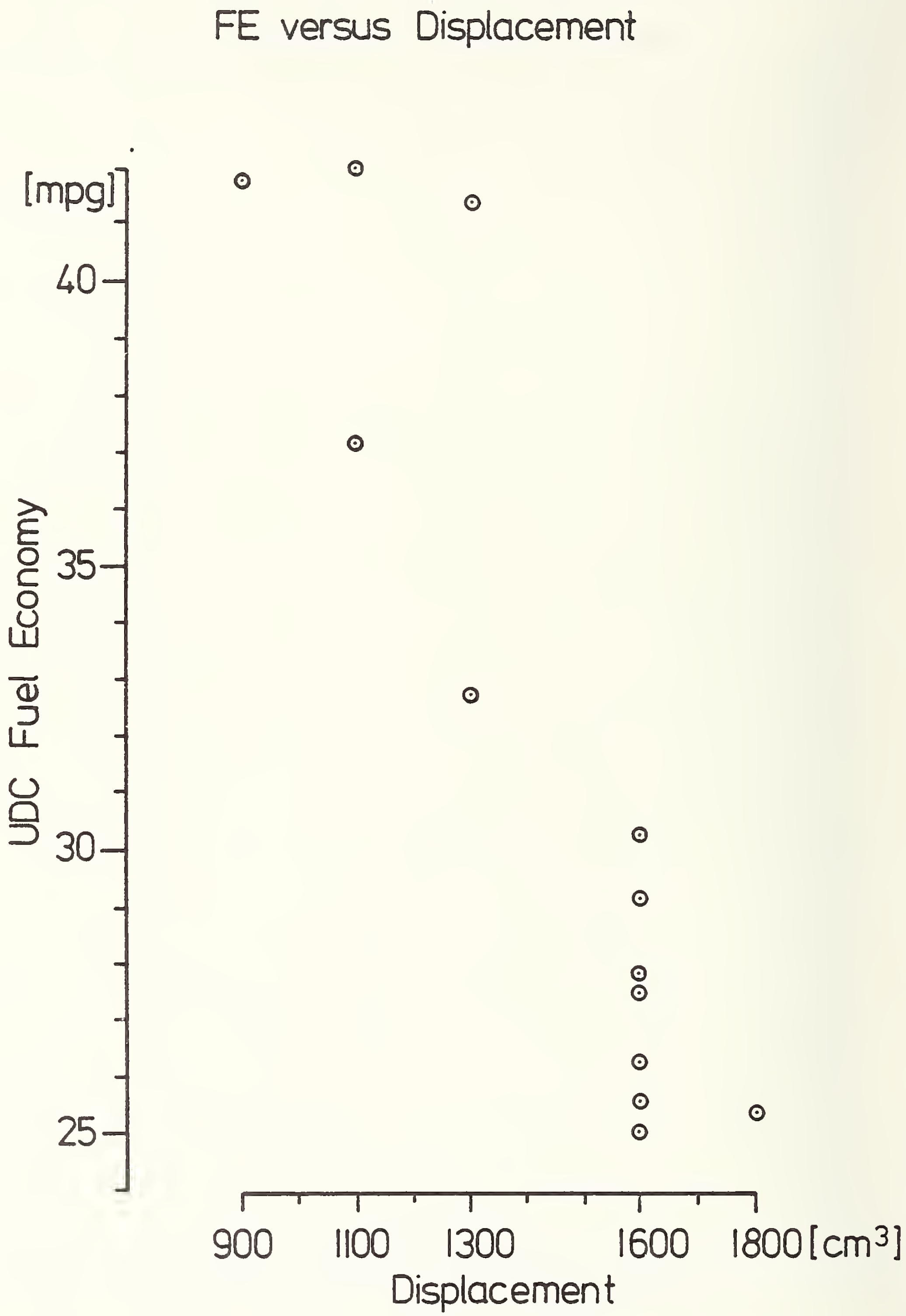

FE versus Displacement
[mpg]
40
35
30
25
UDC Fuel Economy
900
1100
1300
1600
1800 [cm3]
Displacement

Fig.: 6. 1. 60

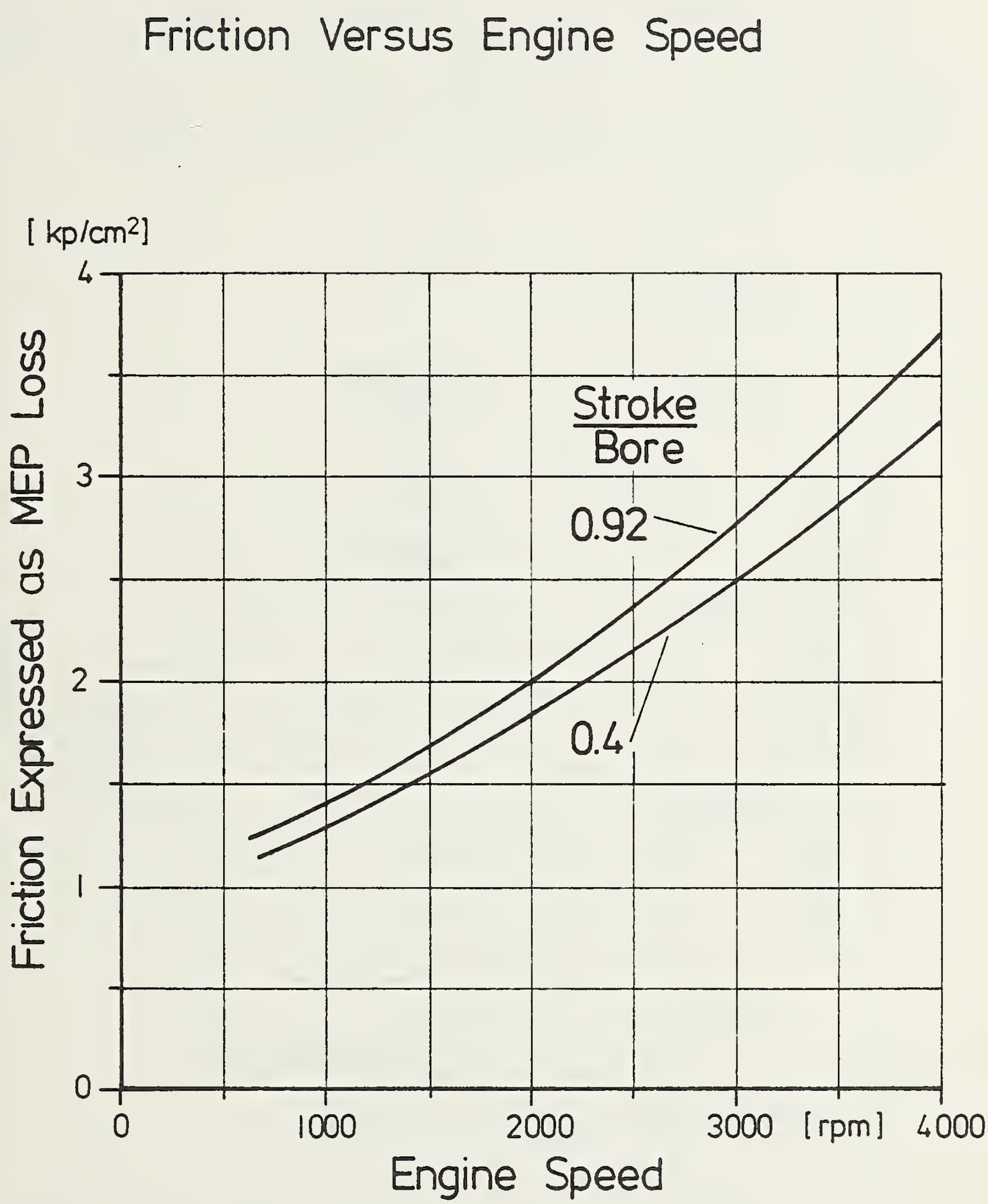

Friction Versus Engine Speed
[kp/cm2]
4
3
2
1
0
Friction Expressed as MEP Loss
Stroke
Bore
0.92
0.4
0
1000
2000
3000 [rpm] 4000
Engine Speed

Fig.: 6. 1. 61 [5]

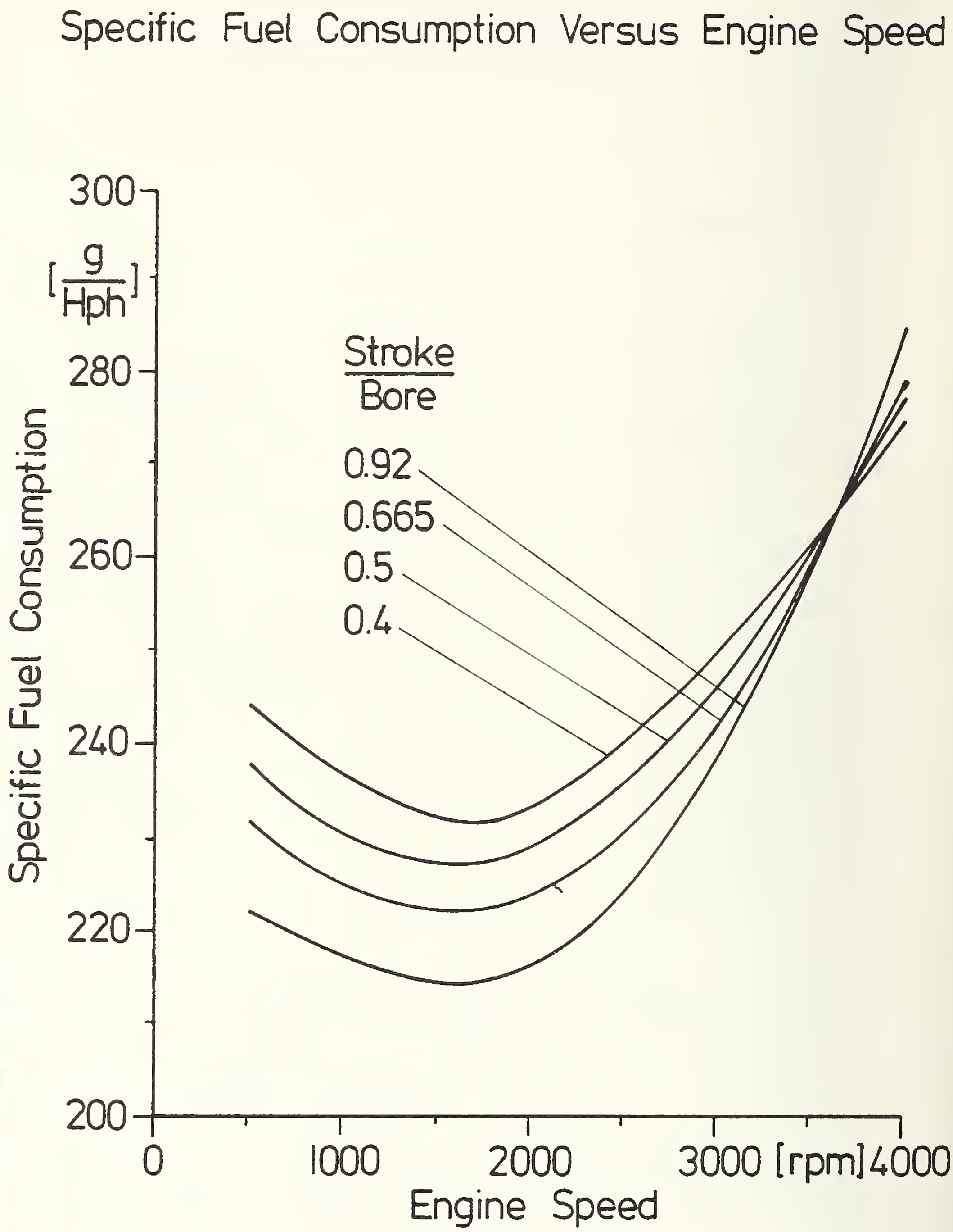

Specific Fuel Consumption Versus Engine Speed
300
[g/Hph]
280
260
240
220
200
Specific Fuel Consumption
Stroke/Bore
0.92
0.665
0.5
0.4
0
1000
2000
3000 [rpm] 4000
Engine Speed

Fig.: 6. 1. 62 [5]

Effect of Piston Speed on Frictional Horse-power (F.H.P.) at Constant Compression Ratio

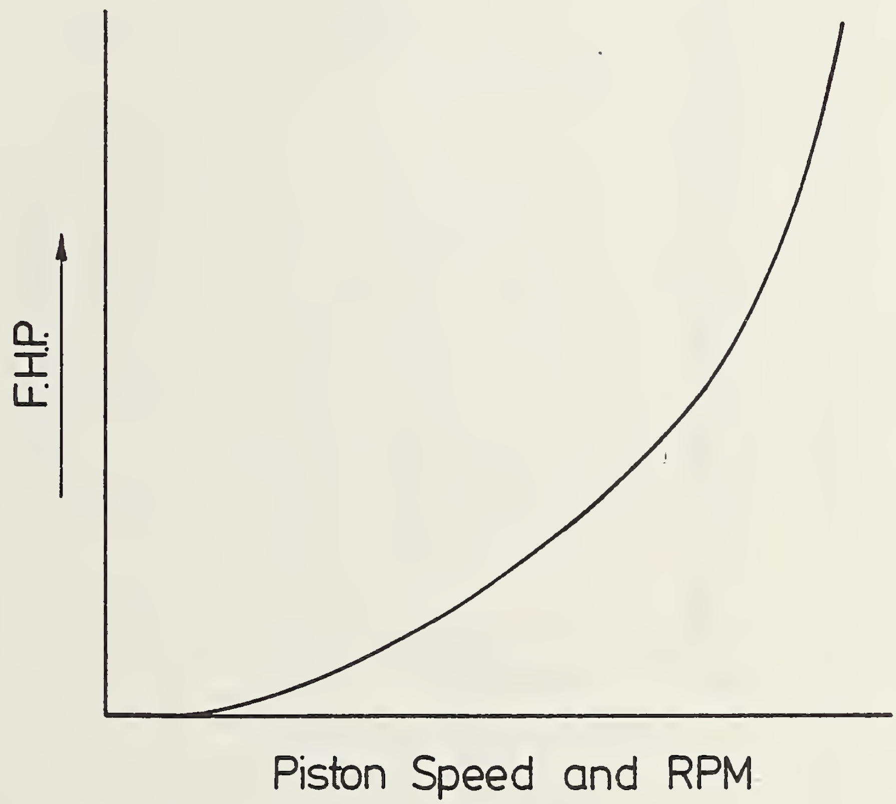


Fig.: 6.1.63 [2]

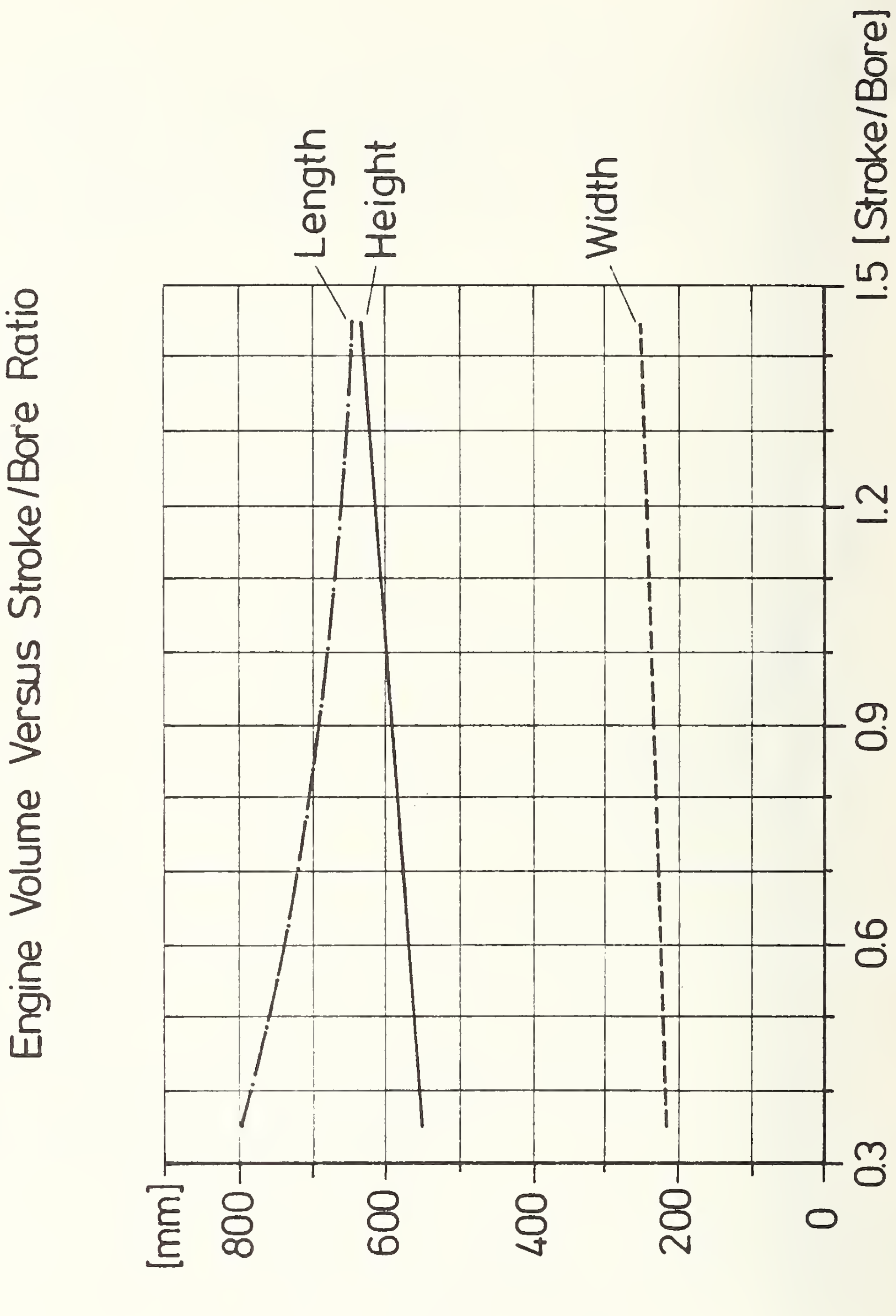
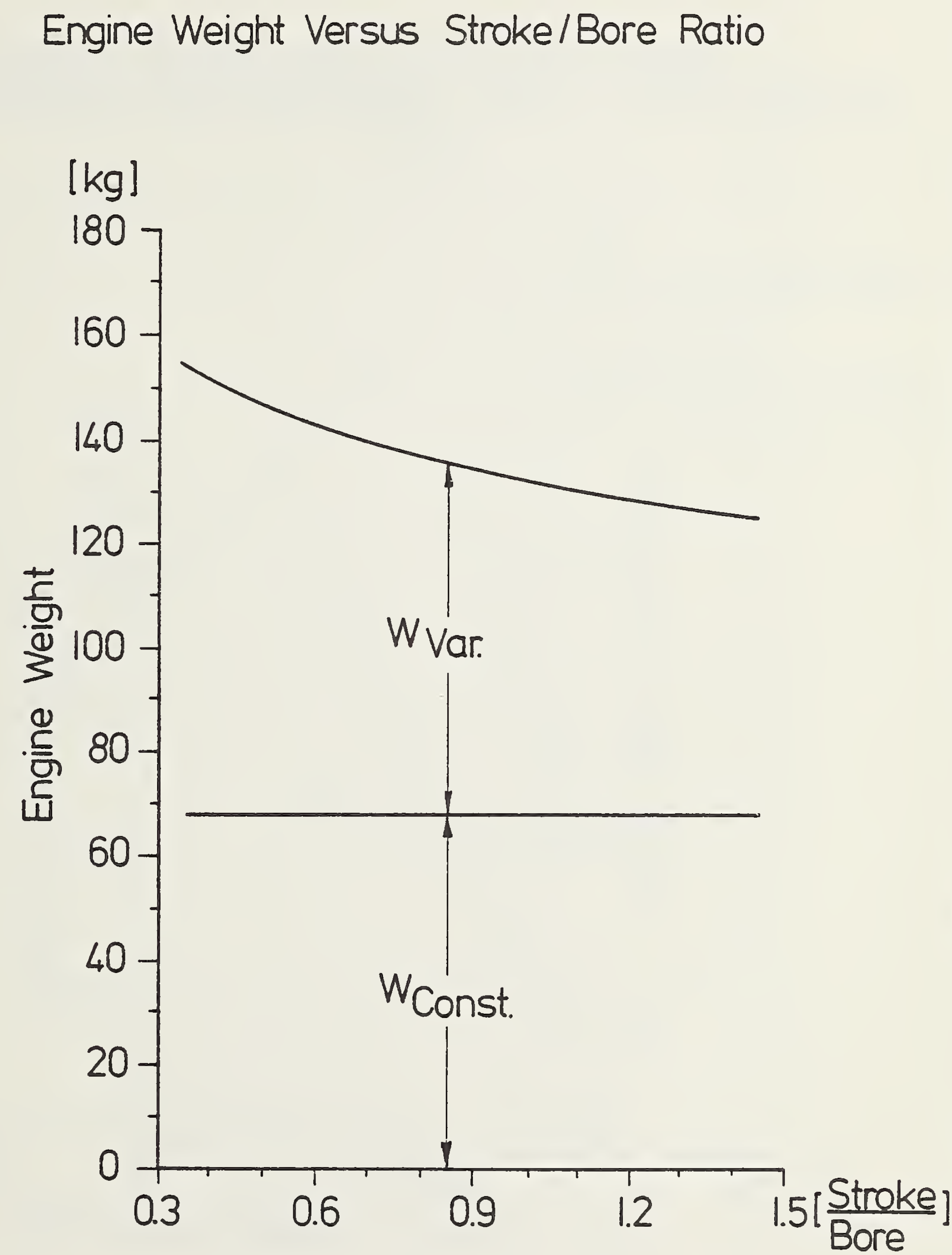

Engine Volume Versus Stroke/Bore Ratio
[mm]
800
600
400
200
0
0.3
0.6
0.9
1.2
1.5 [Stroke/Bore]
Length
Height
Width

Fig.: 6. 1. 64 [5]



Engine Weight Versus Stroke/Bore Ratio
[kg]
180
160
140
120
100
80
60
40
20
0
Engine Weight
W Var.
W Const.
0.3
0.6
0.9
1.2
1.5 [Stroke/Bore]

Fig.: 6. 1.65[5]

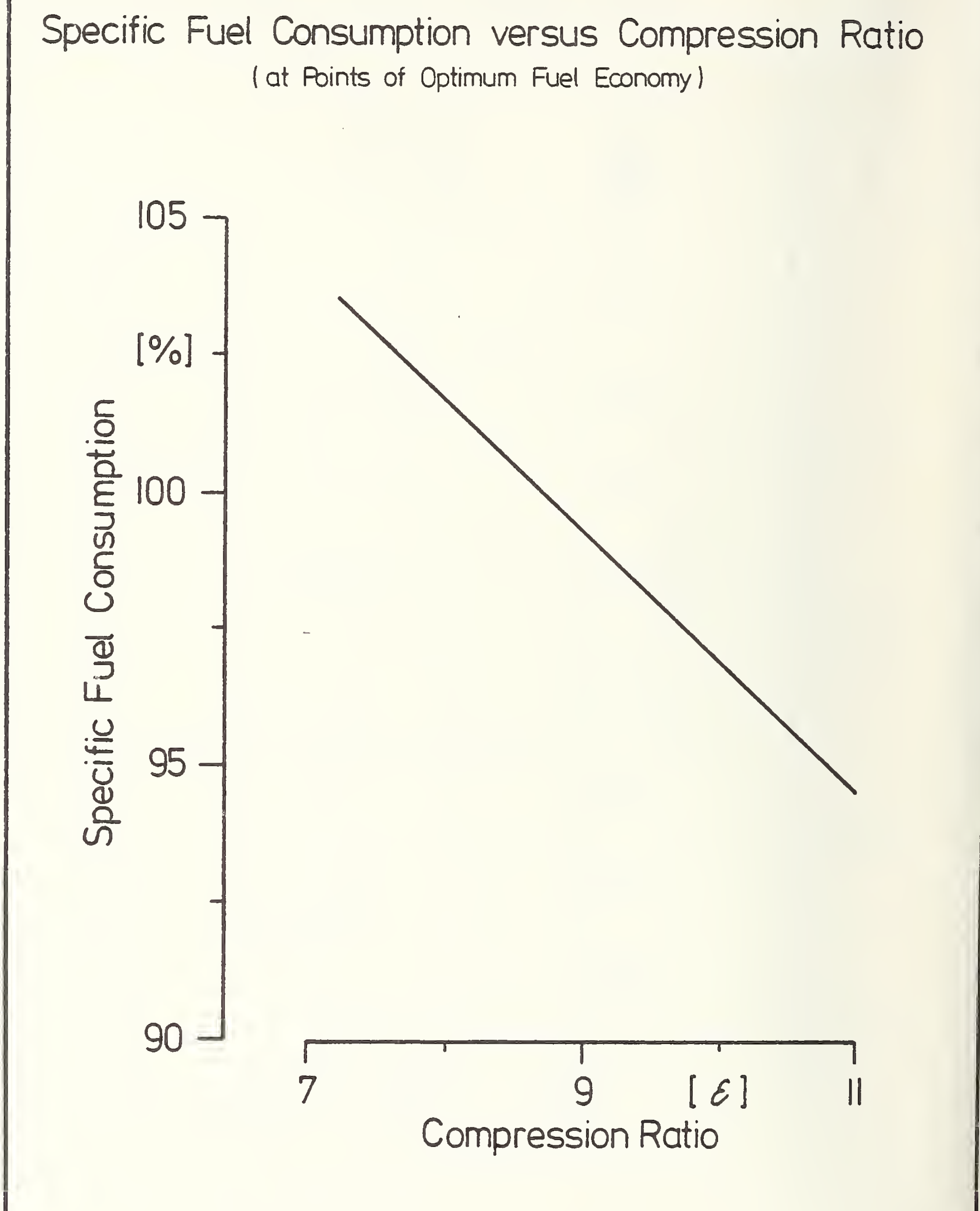

Specific Fuel Consumption versus Compression Ratio
(at Points of Optimum Fuel Economy)
105
[%]
100
95
90
Specific Fuel Consumption
7
9
[ε]
11
Compression Ratio

Fig.: 6. 1. 66

Specific Fuel Consumption versus Throttle Opening Angle

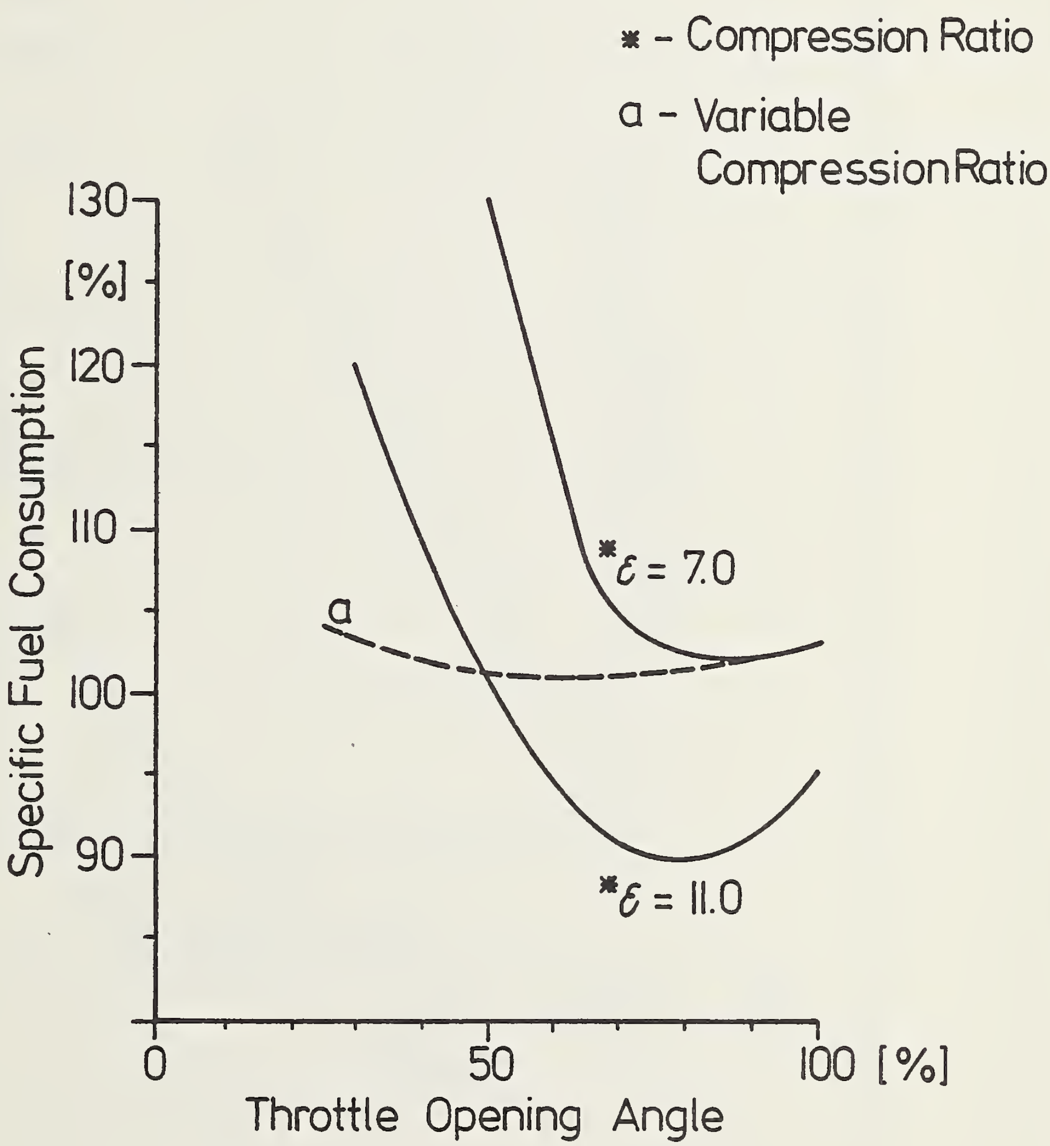


Fig.: 6. I. 67

Knock Intensity versus Throttle Opening

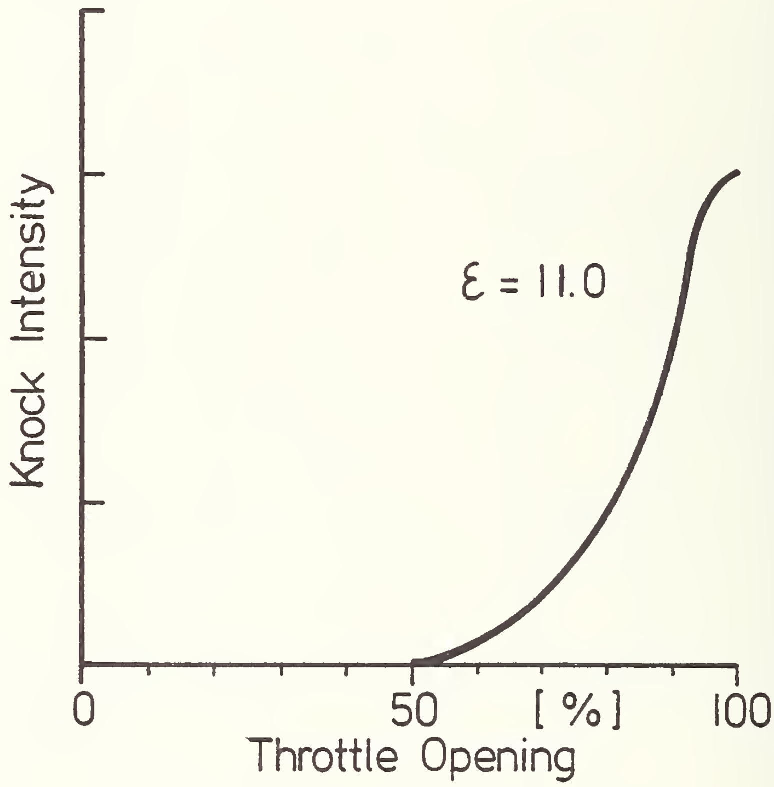


Fig.: 6. 1. 68 [7]

Fuel Economy Advantages Due to Variable Compression Ratio

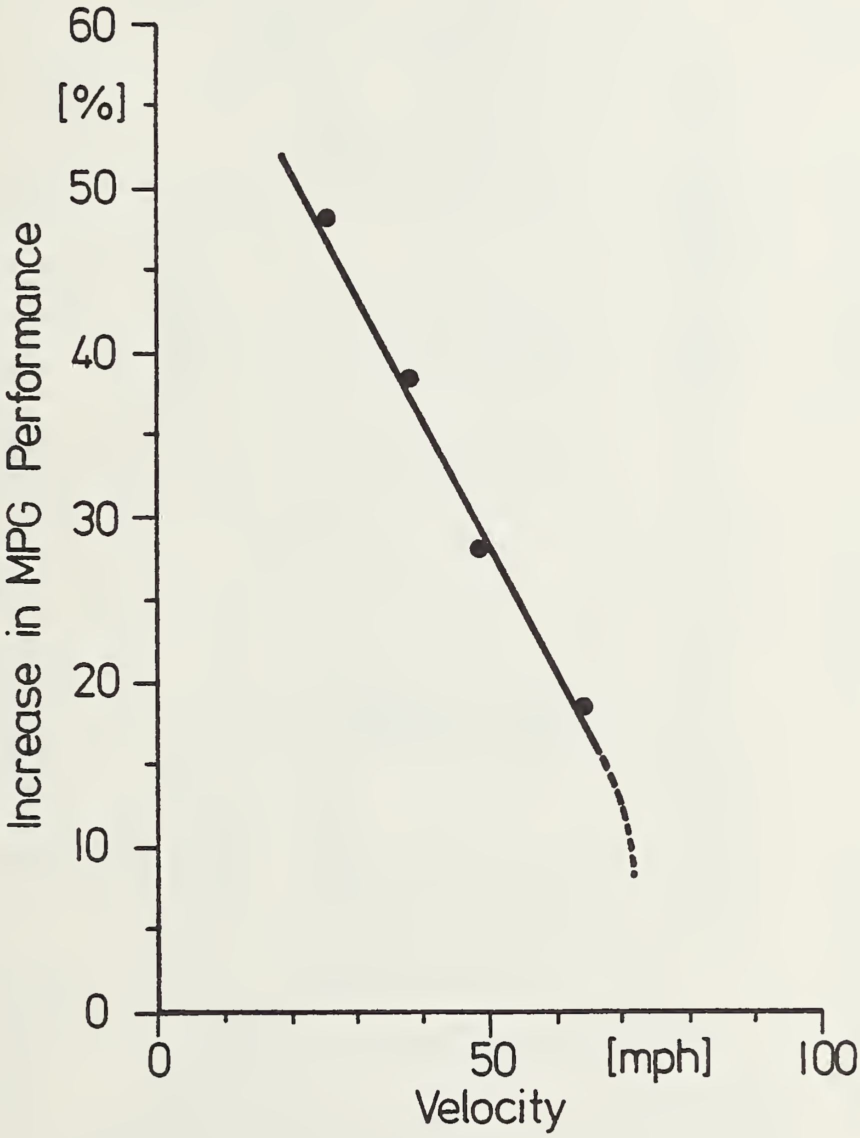


Fig.: 6. I. 69 [7]

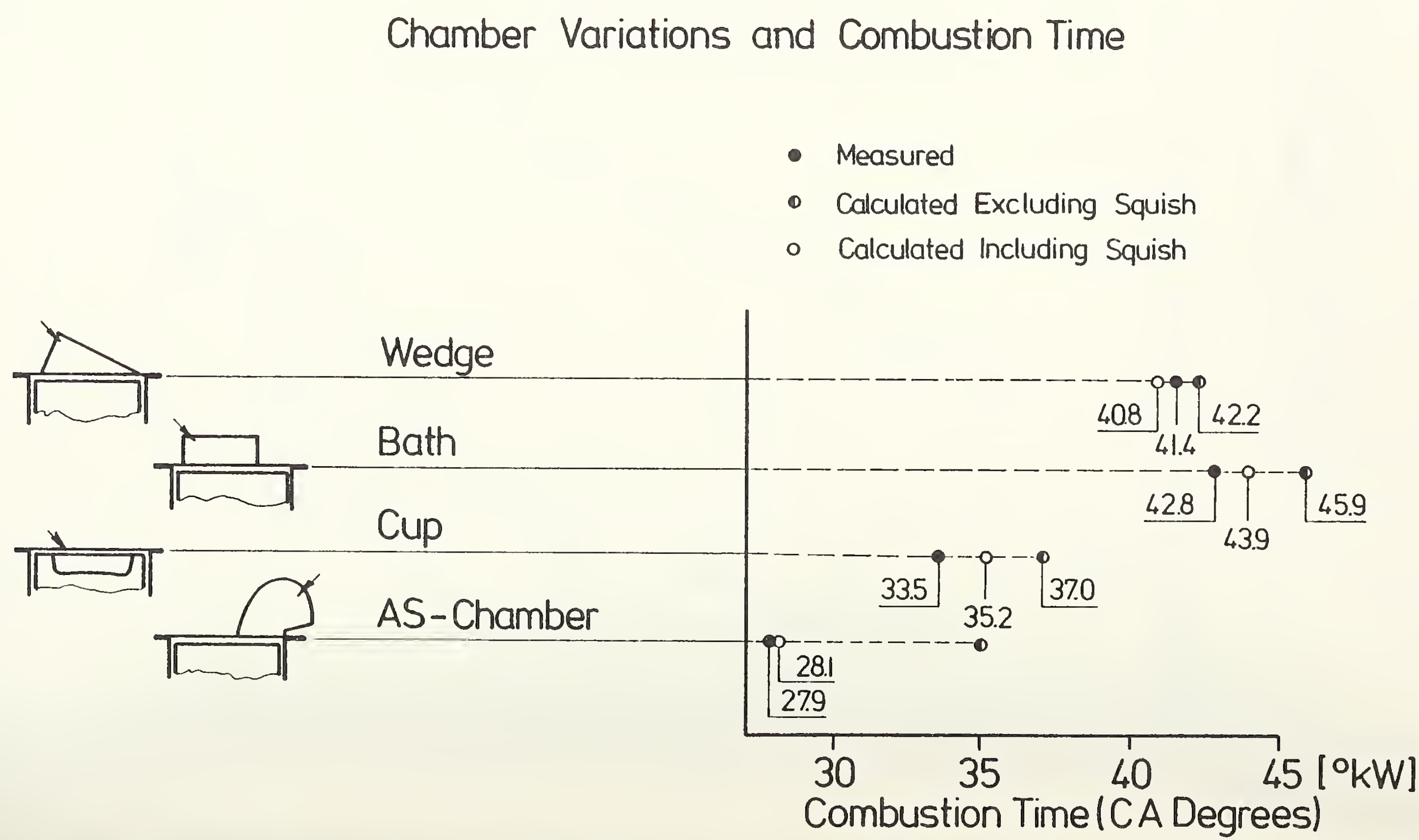
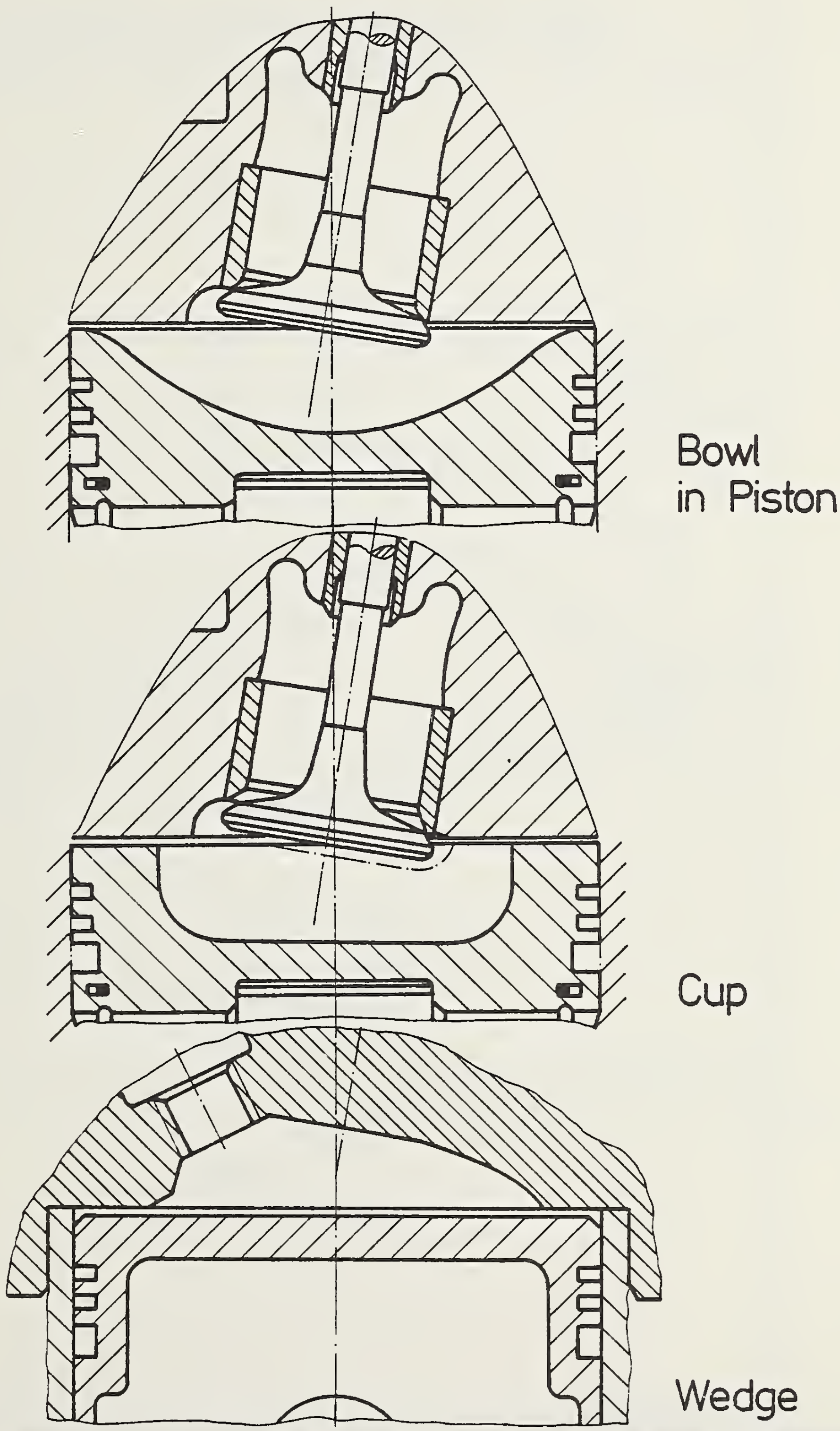
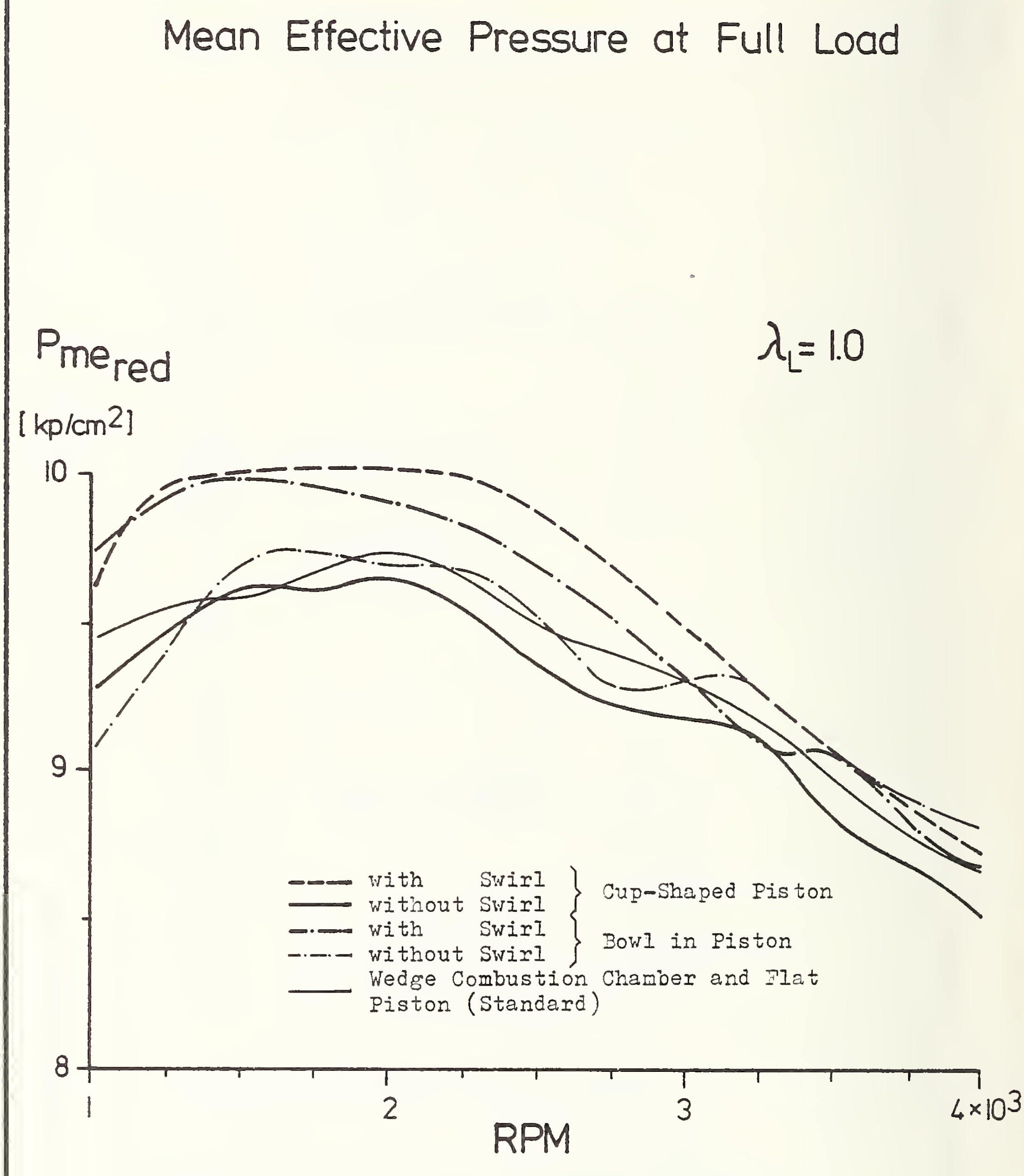

Chamber Variations and Combustion Time
Measured
Calculated Excluding Squish
Calculated Including Squish
Wedge
Bath
Cup
AS-Chamber
40.8
41.4
42.2
42.8
43.9
45.9
33.5
35.2
37.0
28.1
27.9
30
35
40
45 [°kW]
Combustion Time (CA Degrees)

Fig.: 6. 1. 70 [7]

Fig.: 6. 1. 71

VW Combustion Chambers





Mean Effective Pressure at Full Load
$P_{me_{red}}$
[kp/cm2]
$\lambda_L = 1.0$
10
9
8
1
2
3
4×10³
RPM
with Swirl
without Swirl
Cup-Shaped Piston
with Swirl
without Swirl
Bowl in Piston
Wedge Combustion Chamber and Flat Piston (Standard)

Fig.: 6. I. 72

Specific Fuel Consumption at Full Load

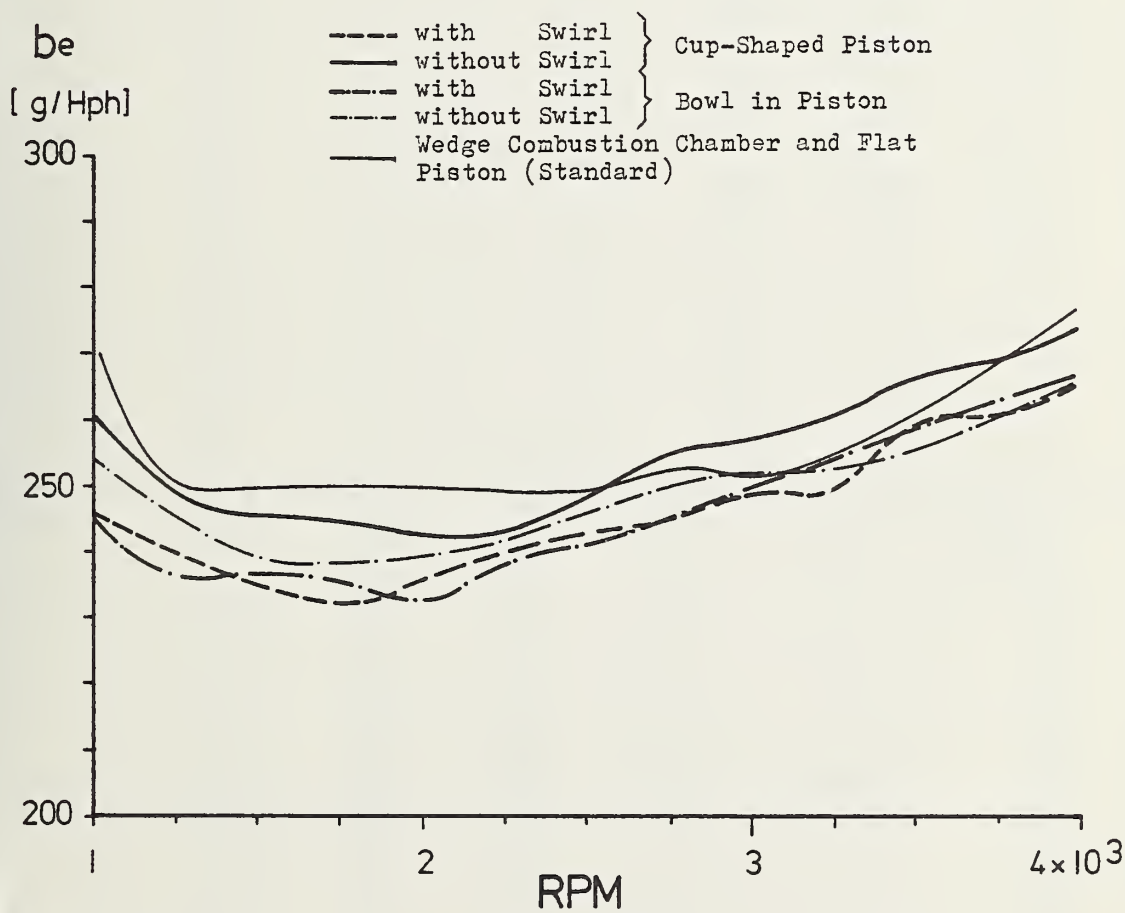


Fig.: 6. I.73

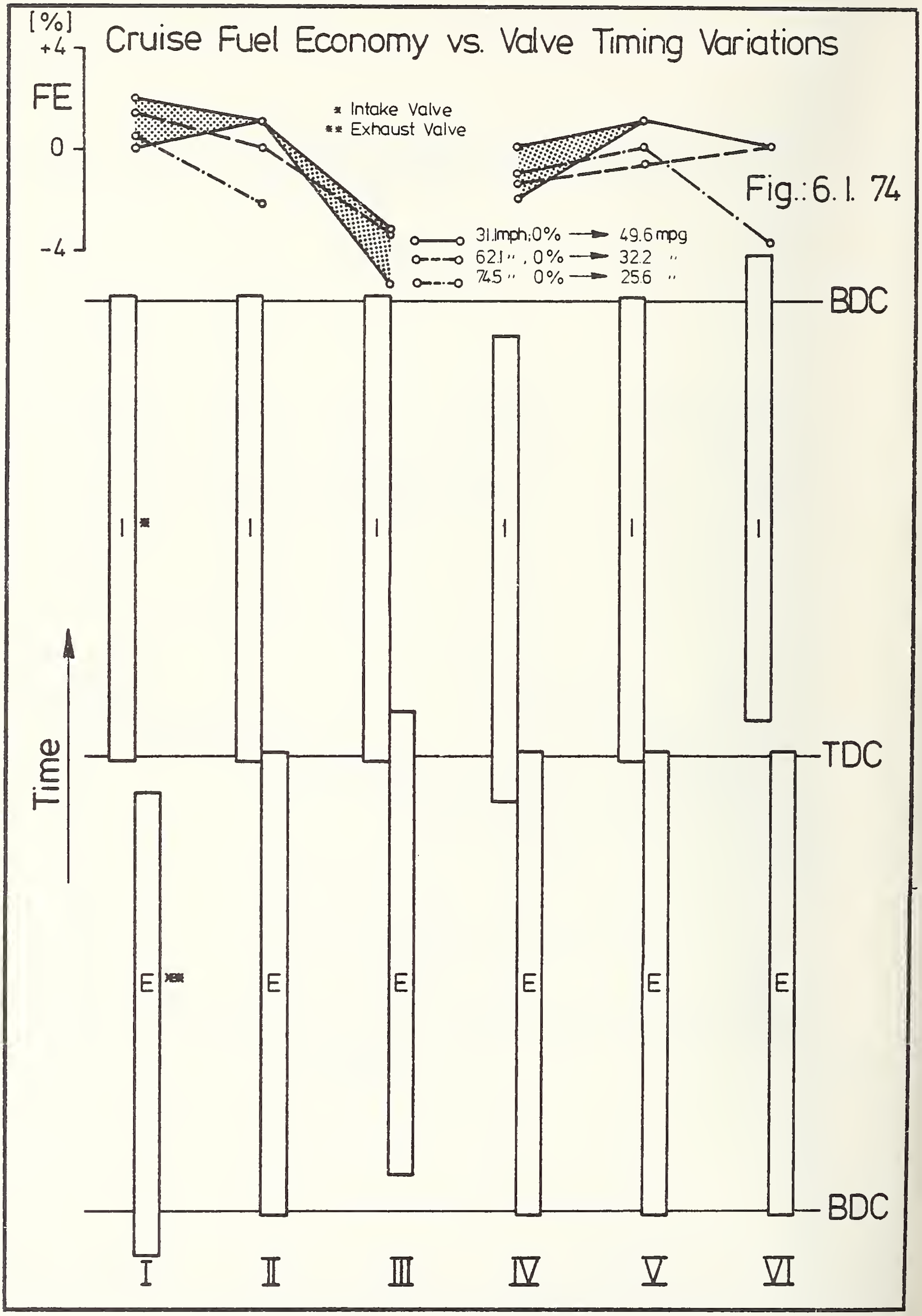
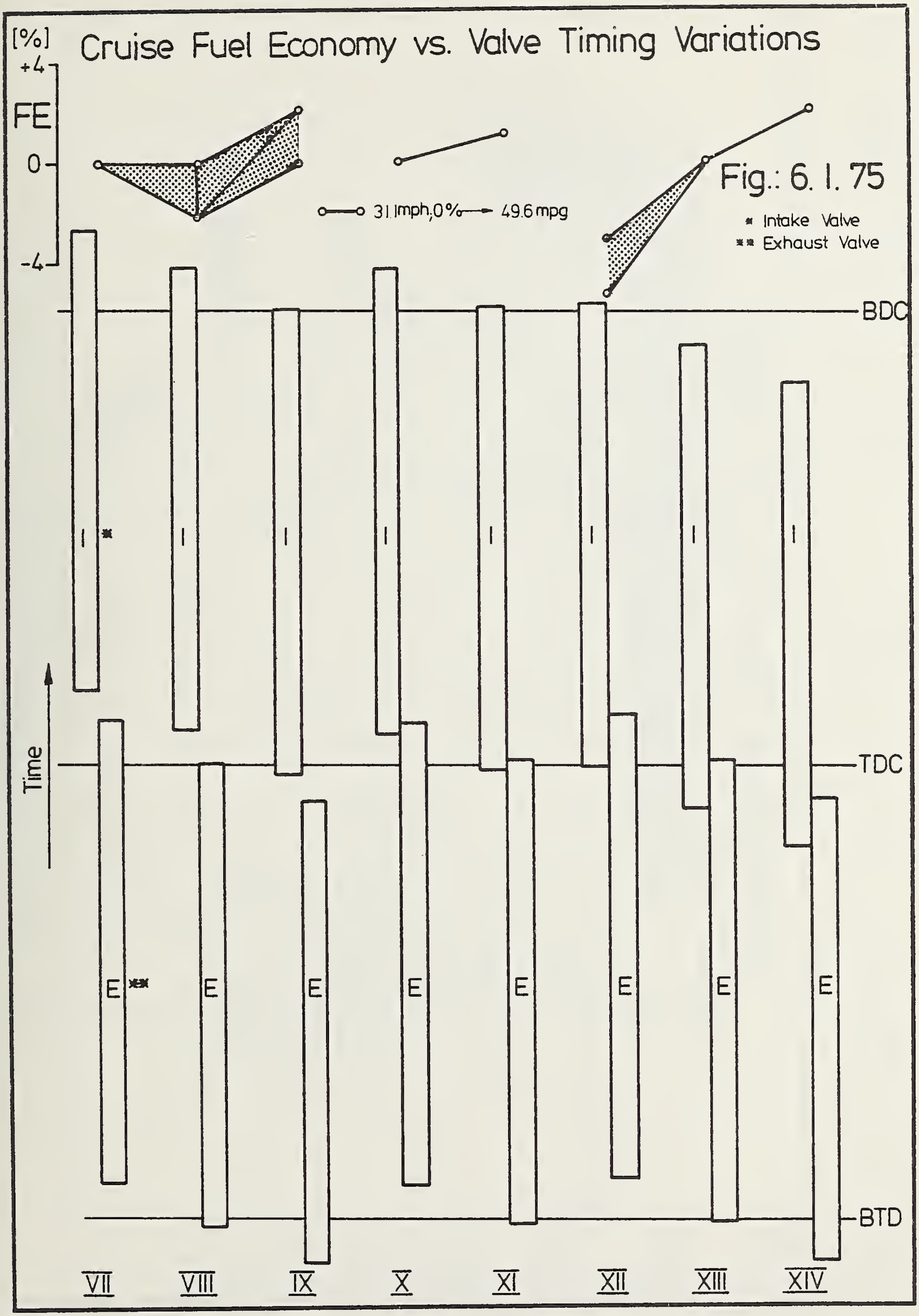

[%]
+4
FE
0
-4
Cruise Fuel Economy vs. Valve Timing Variations
* Intake Valve
** Exhaust Valve
31.1mph; 0% → 49.6 mpg
62.1 " , 0% → 32.2 "
74.5 " 0% → 25.6 "
BDC
TDC
BDC
Time
I *
E **
I
E
I
II
III
IV
V
VI

Fig.: 6.1. 74



[%]
Cruise Fuel Economy vs. Valve Timing Variations
+4
FE
0
-4
31.1mph,0%—49.6mpg
Fig.: 6.1.75
* Intake Valve
** Exhaust Valve
BDC
TDC
BTD
Time
I
E
VII
VIII
IX
X
XI
XII
XIII
XIV

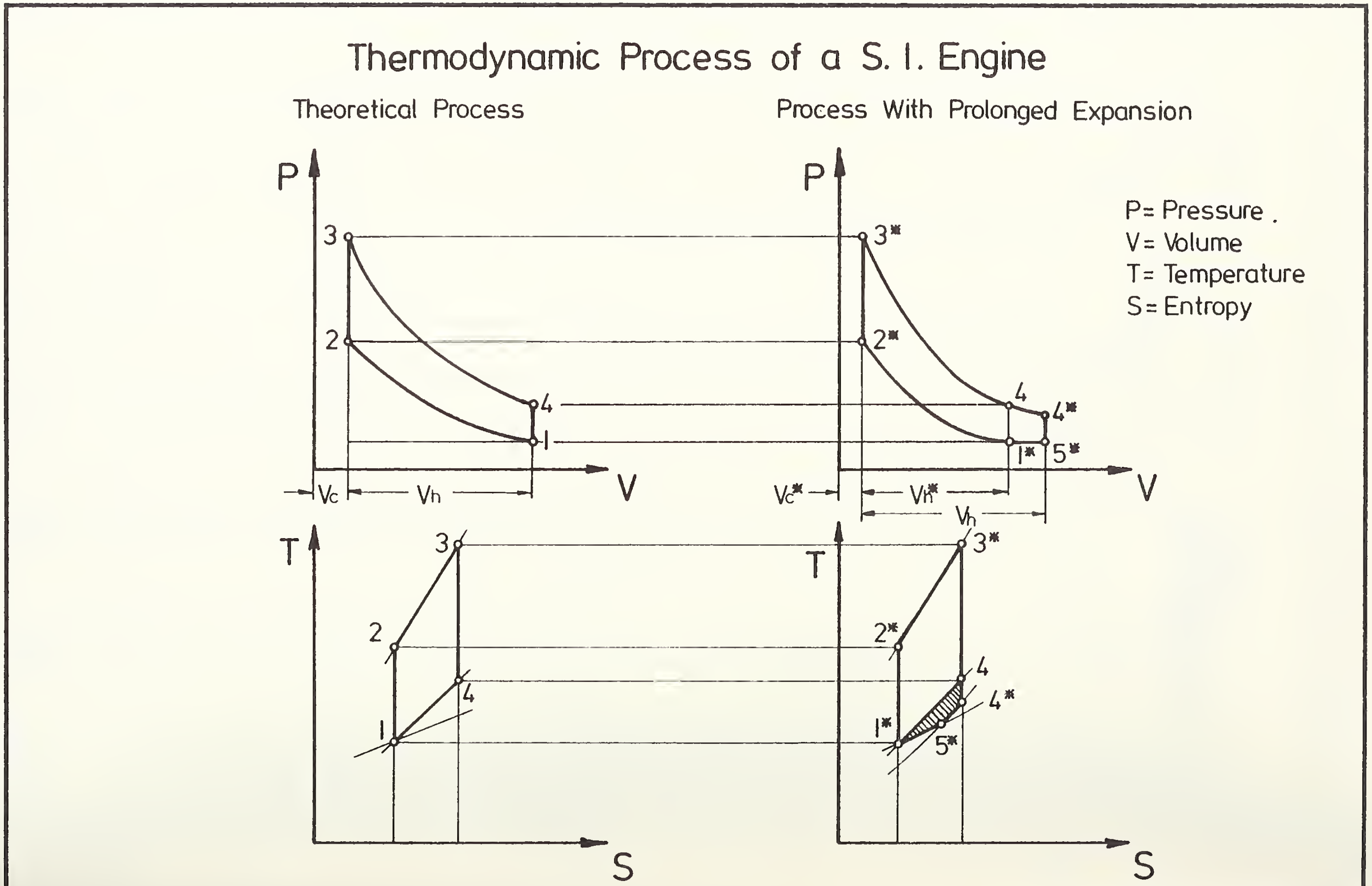

Thermodynamic Process of a S. I. Engine
Theoretical Process
Process With Prolonged Expansion
P= Pressure .
V= Volume
T= Temperature
S= Entropy
P
V
T
S
Vc
Vh
Vc*
Vh*
1
2
3
4
1*
2*
3*
4*
5*

Fig.: 6. I. 76[8]

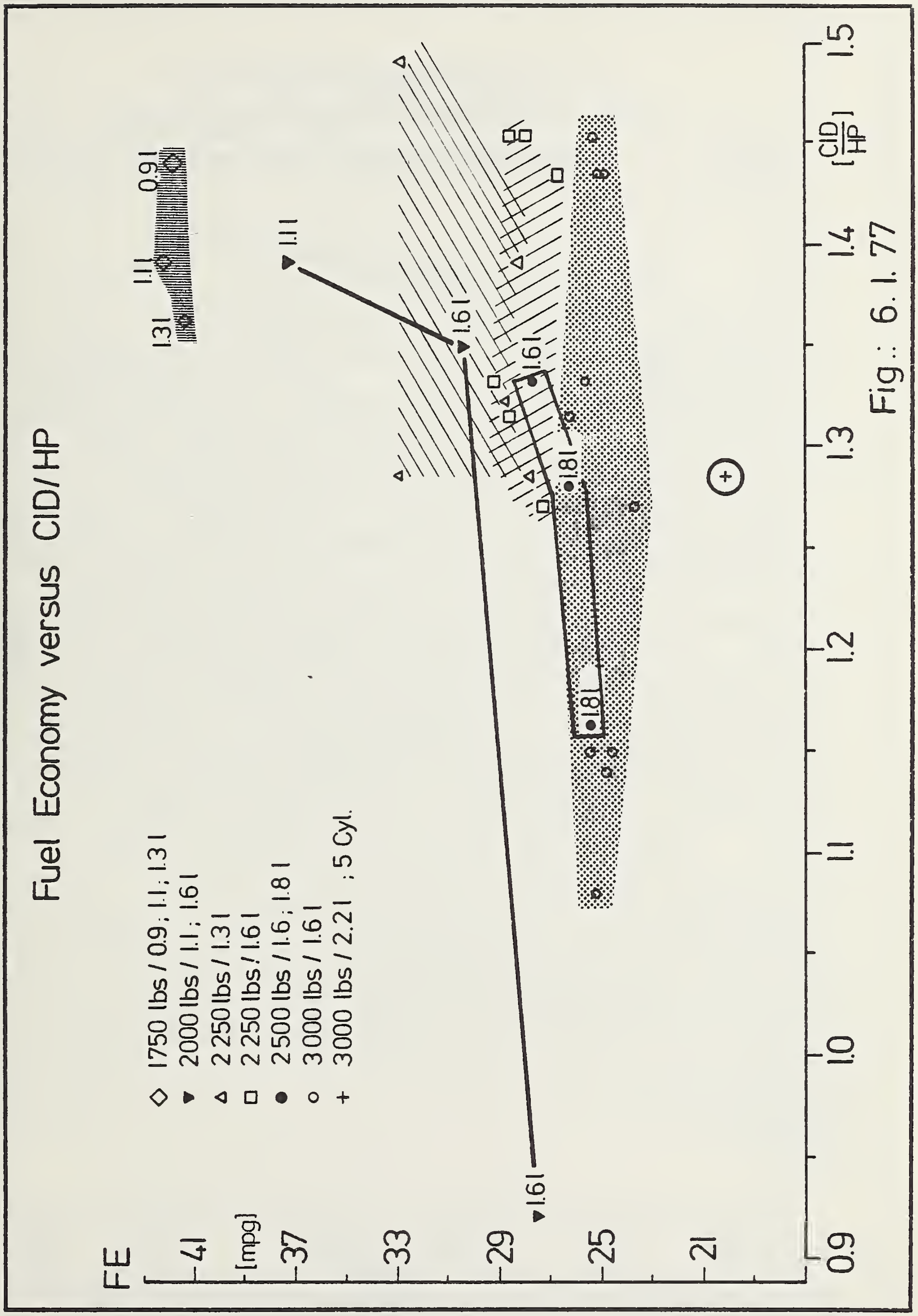

Fuel Economy versus CID/HP
◇ 1750 lbs / 0.9; 1.1; 1.3 l
▼ 2000 lbs / 1.1; 1.6 l
△ 2250 lbs / 1.3 l
□ 2250 lbs / 1.6 l
● 2500 lbs / 1.6; 1.8 l
○ 3000 lbs / 1.6 l
+ 3000 lbs / 2.2 l ; 5 Cyl.
FE
[mpg]
41
37
33
29
25
21
0.9
1.0
1.1
1.2
1.3
1.4
1.5
[CID/HP]
1.6 l
1.1 l
1.3 l
0.9 l
1.8 l

Fig.: 6. 1. 77

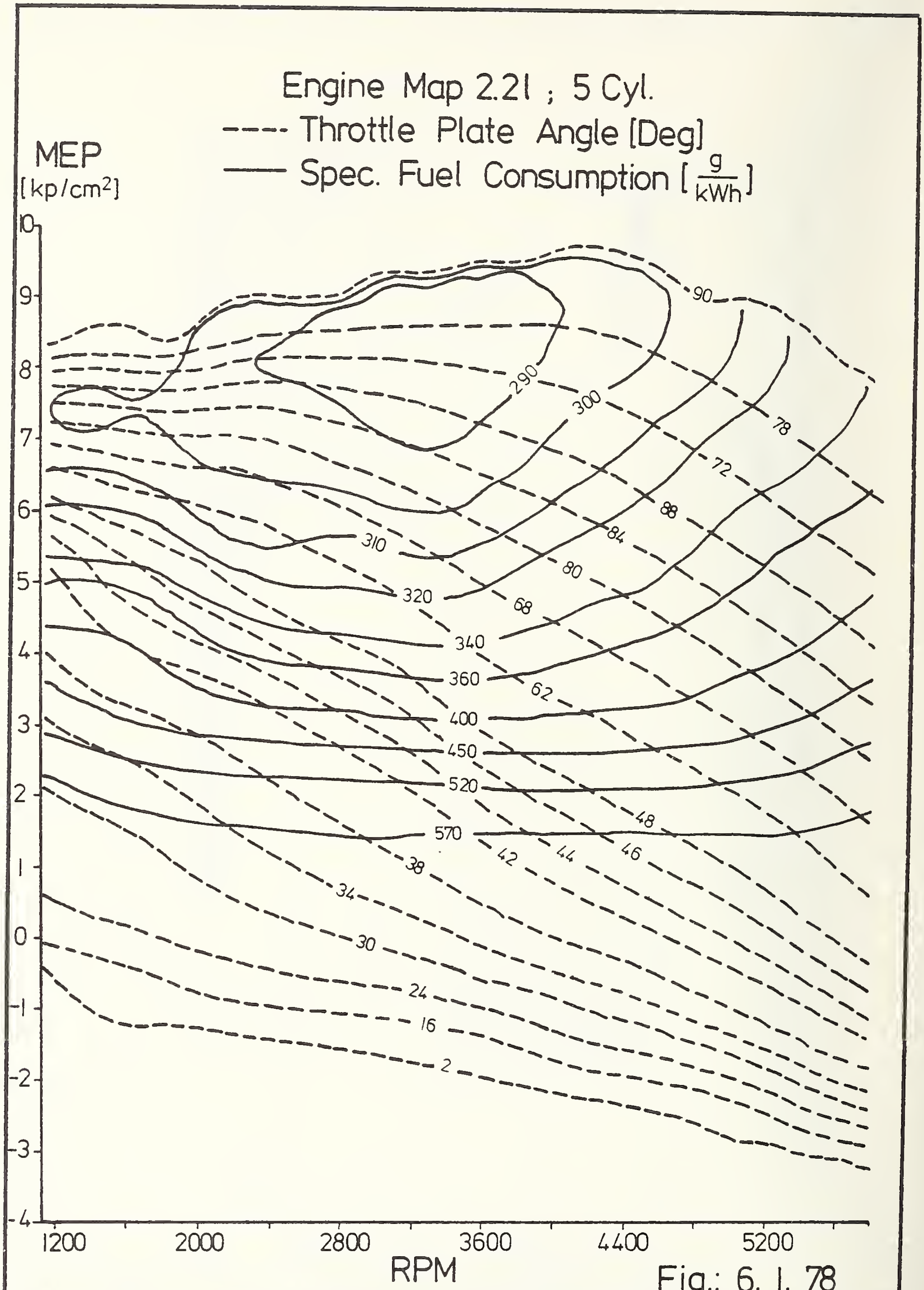

Engine Map 2.2l ; 5 Cyl.
---- Throttle Plate Angle [Deg]
—— Spec. Fuel Consumption [g/kWh]
MEP [kp/cm2]
10
9
8
7
6
5
4
3
2
1
0
-1
-2
-3
-4
1200
2000
2800
3600
4400
5200
RPM
90
78
72
88
84
80
68
62
48
46
44
42
38
34
30
24
16
2
290
300
310
320
340
360
400
450
520
570

Fig.: 6. l. 78

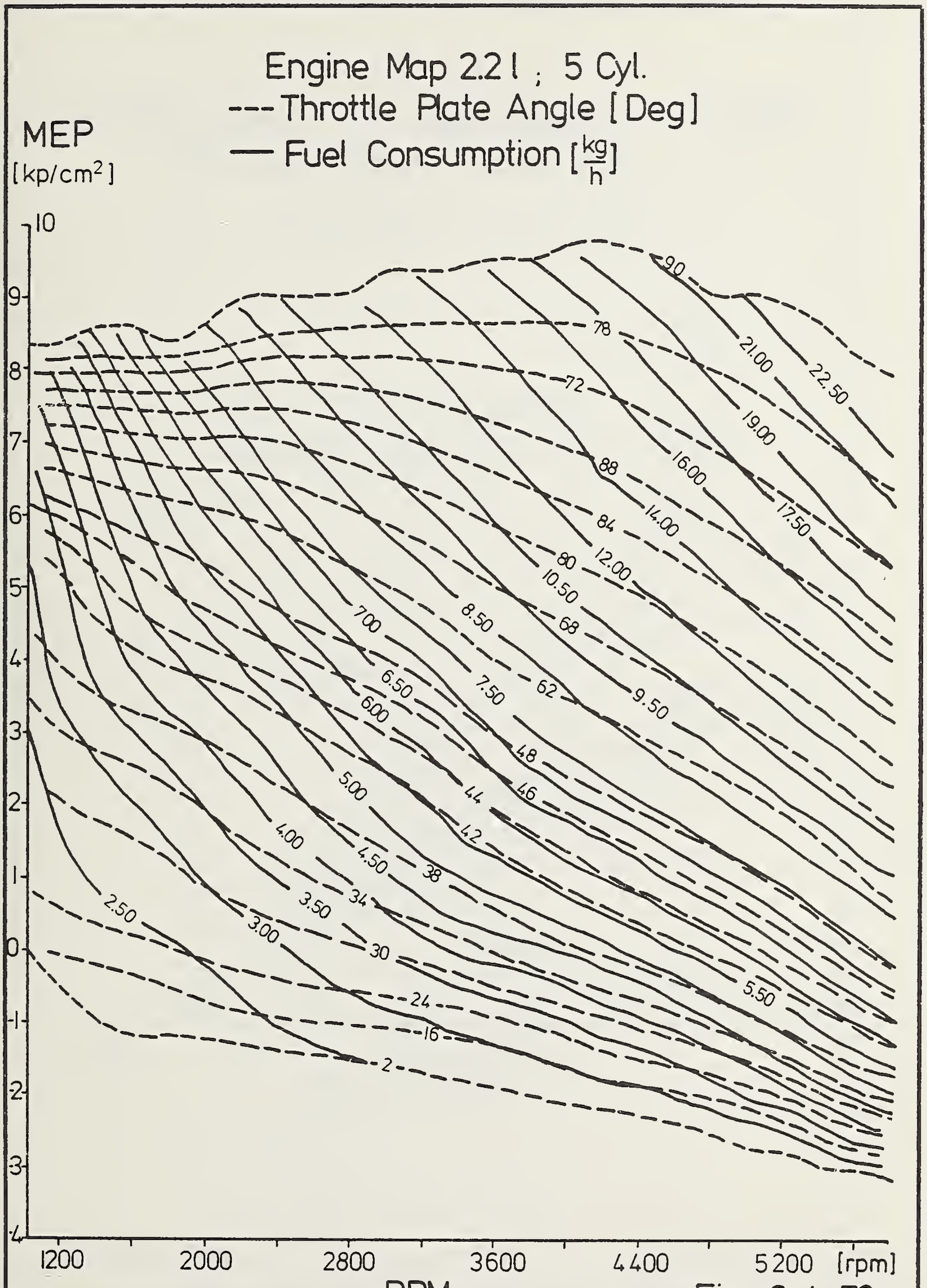

Engine Map 2.2 l ; 5 Cyl.
--- Throttle Plate Angle [Deg]
— Fuel Consumption [kg/h]
MEP
[kp/cm²]
10
9
8
7
6
5
4
3
2
1
0
-1
-2
-3
-4
1200
2000
2800
3600
4400
5200
[rpm]
RPM
90
78
72
88
84
80
68
62
48
46
44
42
38
34
30
24
16
2
22.50
21.00
19.00
17.50
16.00
14.00
12.00
10.50
9.50
8.50
7.50
7.00
6.50
6.00
5.50
5.00
4.50
4.00
3.50
3.00
2.50

Fig.: 6. I. 79

Effect of Air/Fuel Ratio on Thermal Efficiency η_i
at Constant Compression Ratio

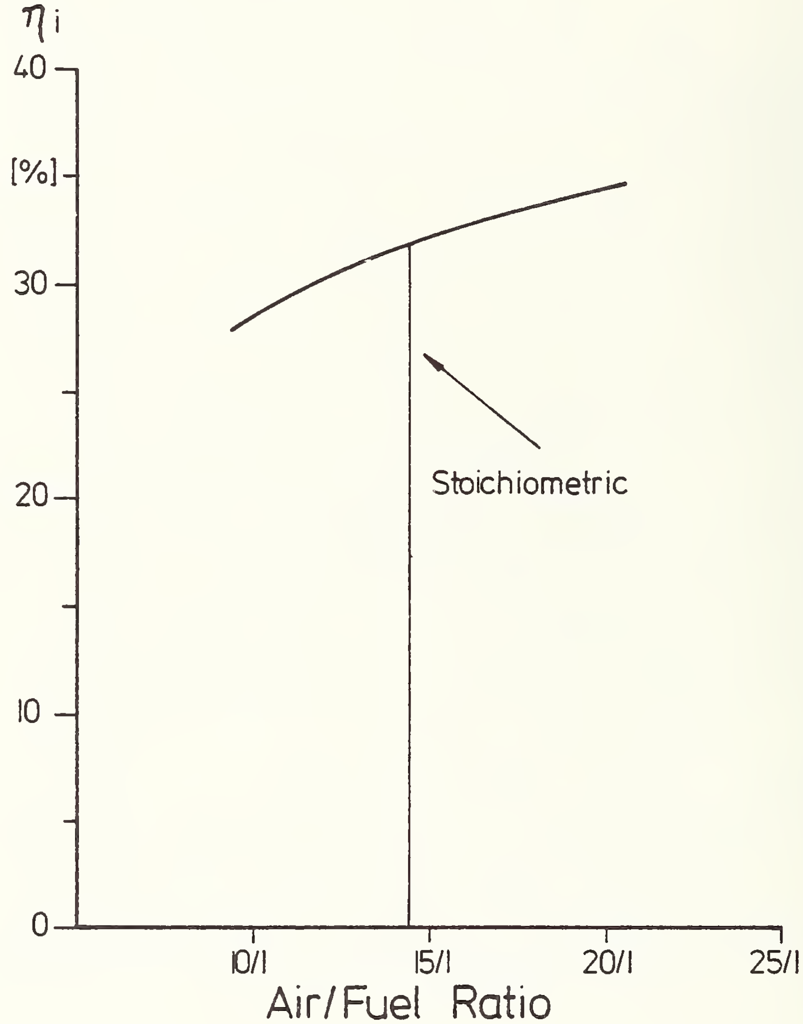


Fig.: 6. 1. 80 [2]

Fuel Flow vs. Air Flow

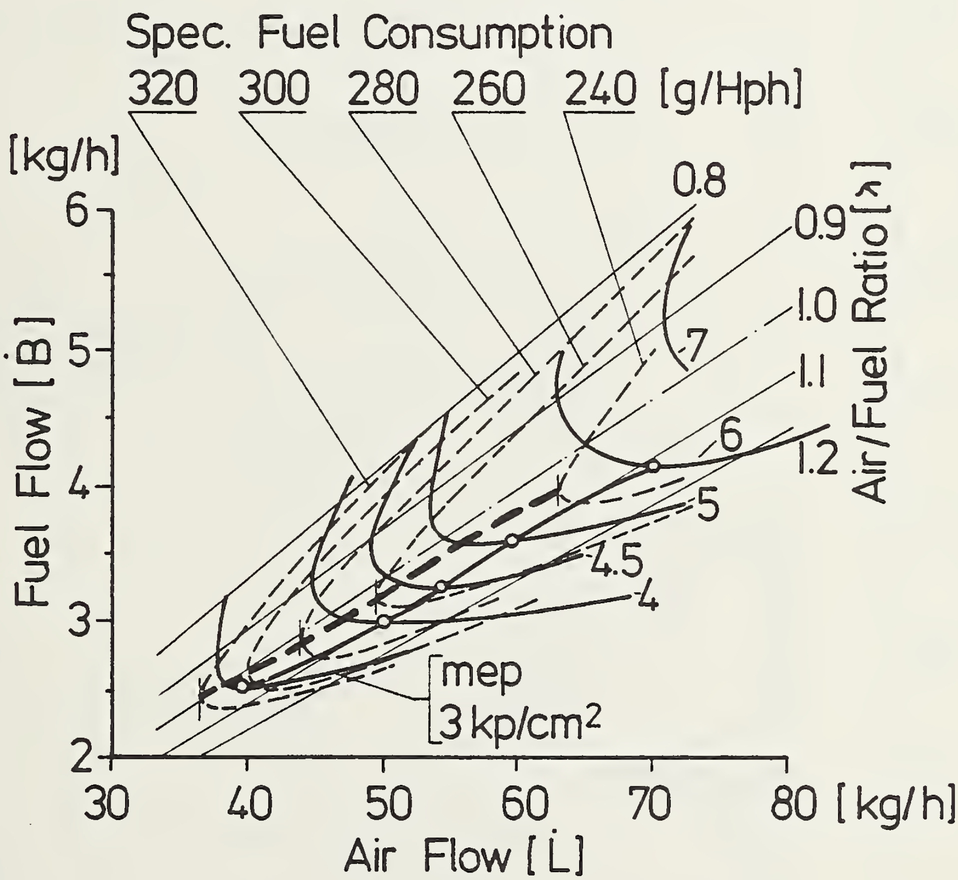
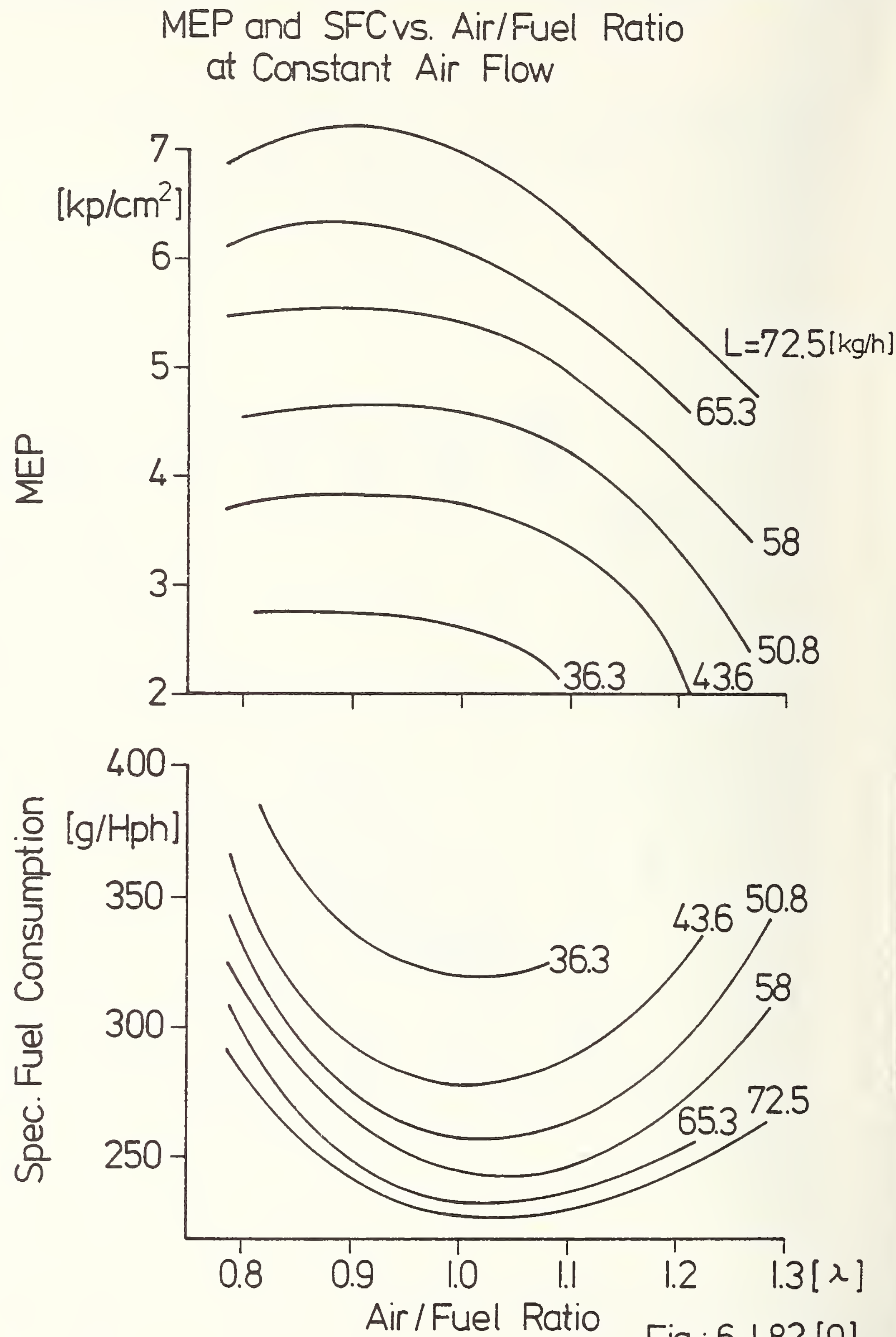
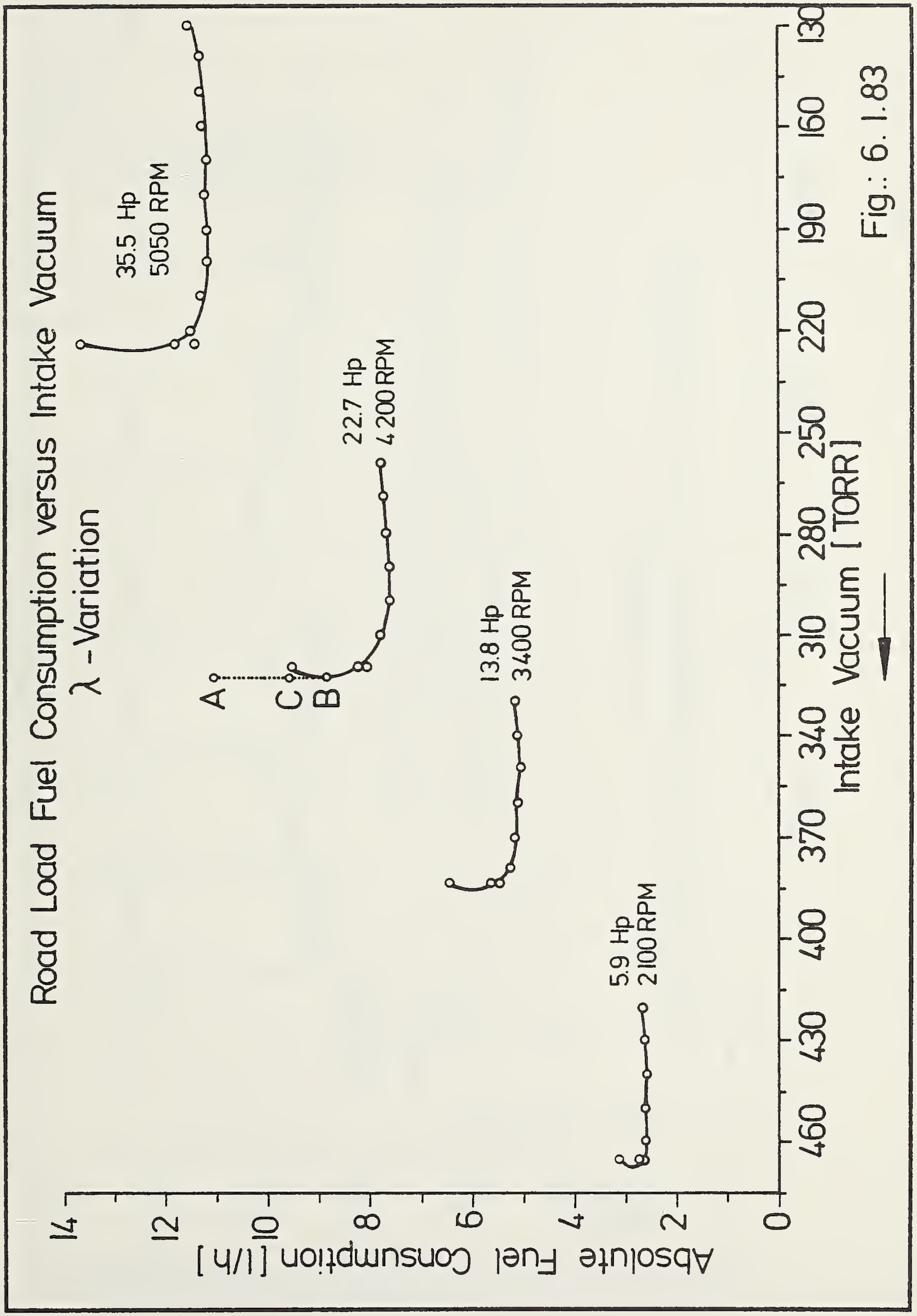


Fig.: 6. 1.81 [9]



MEP and SFC vs. Air/Fuel Ratio
at Constant Air Flow
7
[kp/cm²]
6
5
4
3
2
MEP
L=72.5 [kg/h]
65.3
58
50.8
43.6
36.3
400
[g/Hph]
350
300
250
Spec. Fuel Consumption
36.3
43.6
50.8
58
65.3
72.5
0.8
0.9
1.0
1.1
1.2
1.3 [λ]
Air / Fuel Ratio

Fig.: 6.1.82 [9]



Road Load Fuel Consumption versus Intake Vacuum
λ -Variation
35.5 Hp
5050 RPM
22.7 Hp
4200 RPM
13.8 Hp
3400 RPM
5.9 Hp
2100 RPM
A
C
B
Absolute Fuel Consumption [l/h]
Intake Vacuum [TORR]
0
2
4
6
8
10
12
14
460
430
400
370
340
310
280
250
220
190
160
130

Fig.: 6.1.83

Full Load Fuel Consumption versus Horsepower

λ - Variation

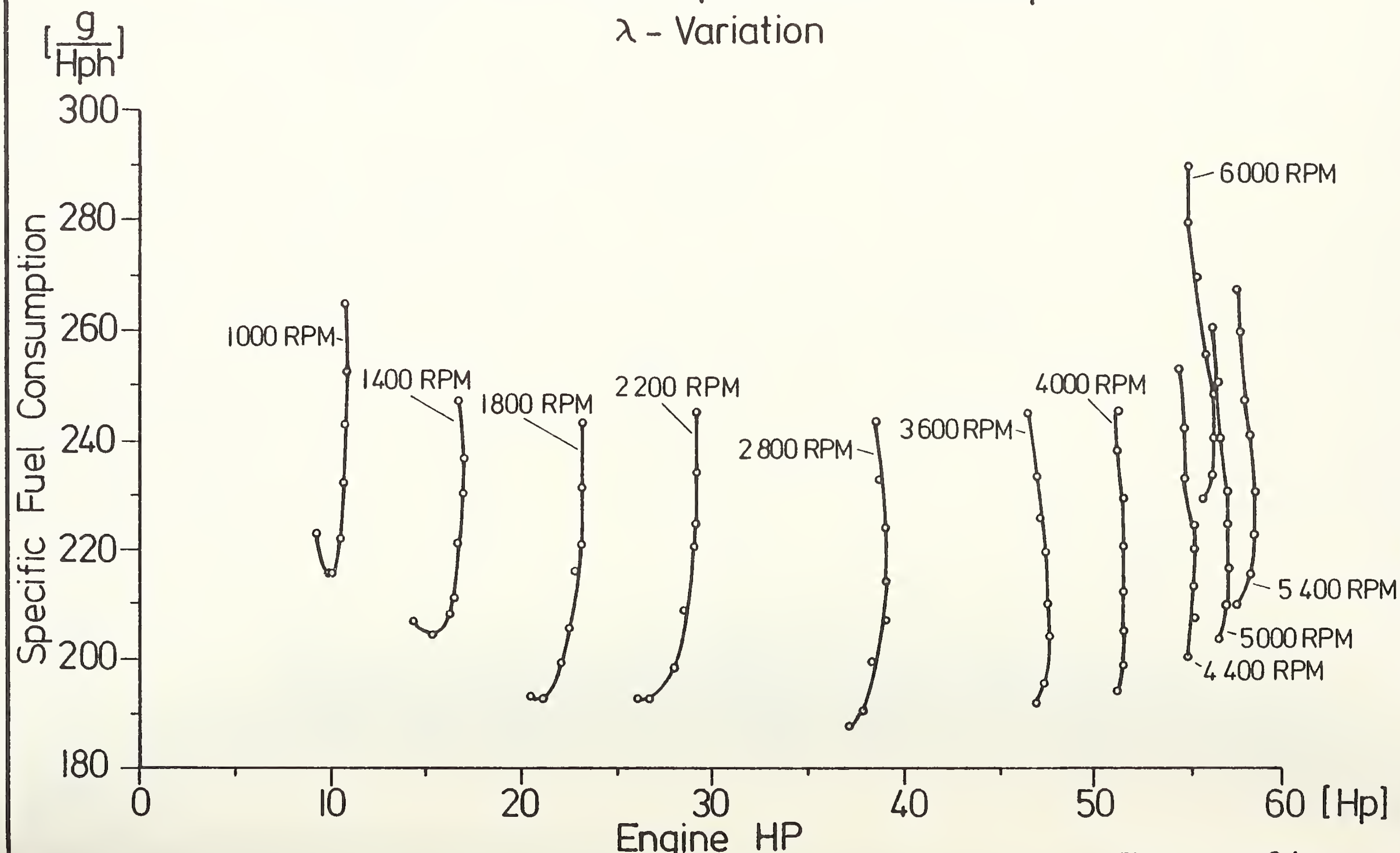
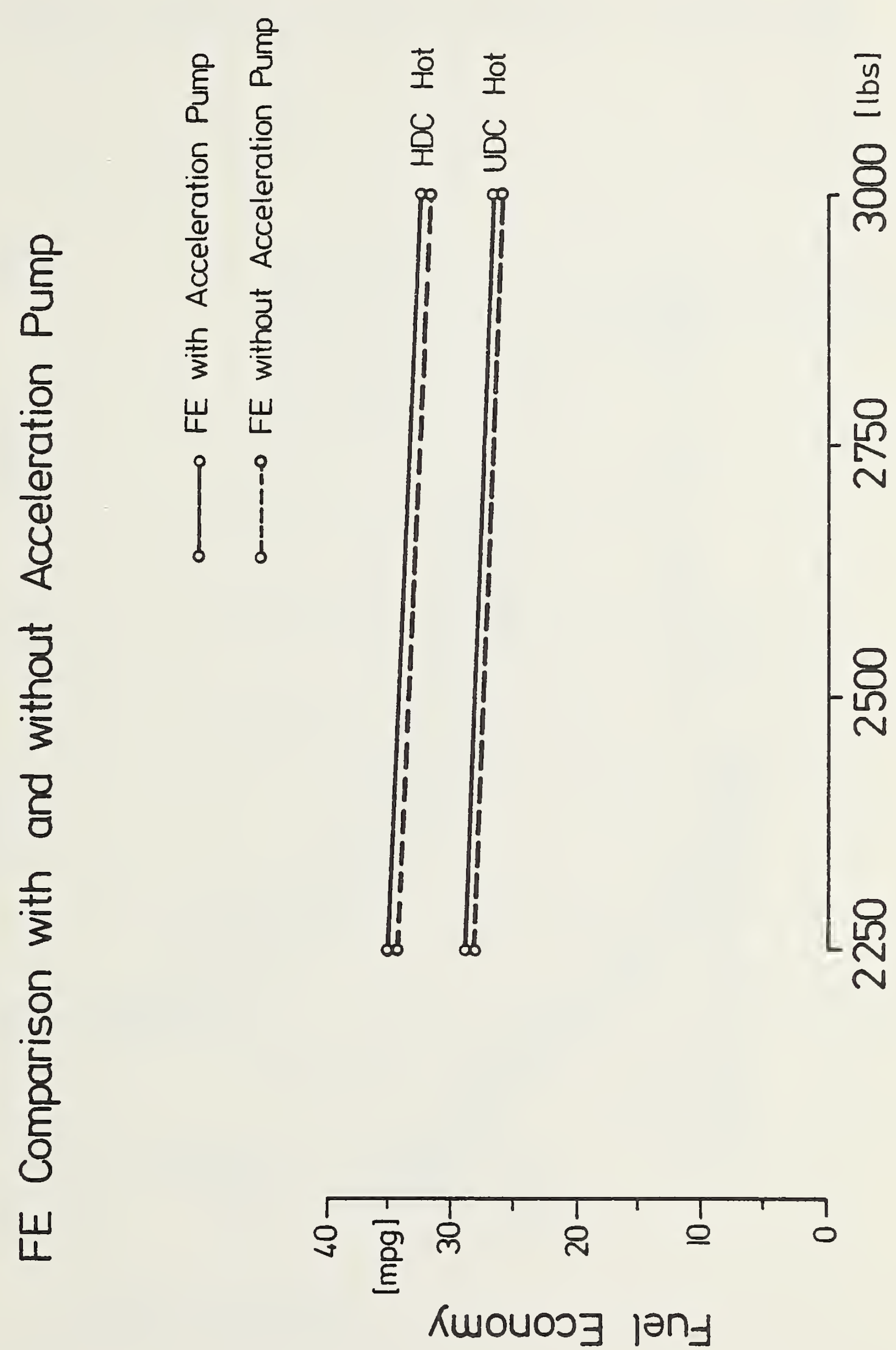


Fig.: 6. 1. 84



FE Comparison with and without Acceleration Pump
FE with Acceleration Pump
FE without Acceleration Pump
HDC Hot
UDC Hot
Fuel Economy
[mpg]
40
30
20
10
0
2250
2500
2750
3000
[lbs]

Fig.: 6. I. 85

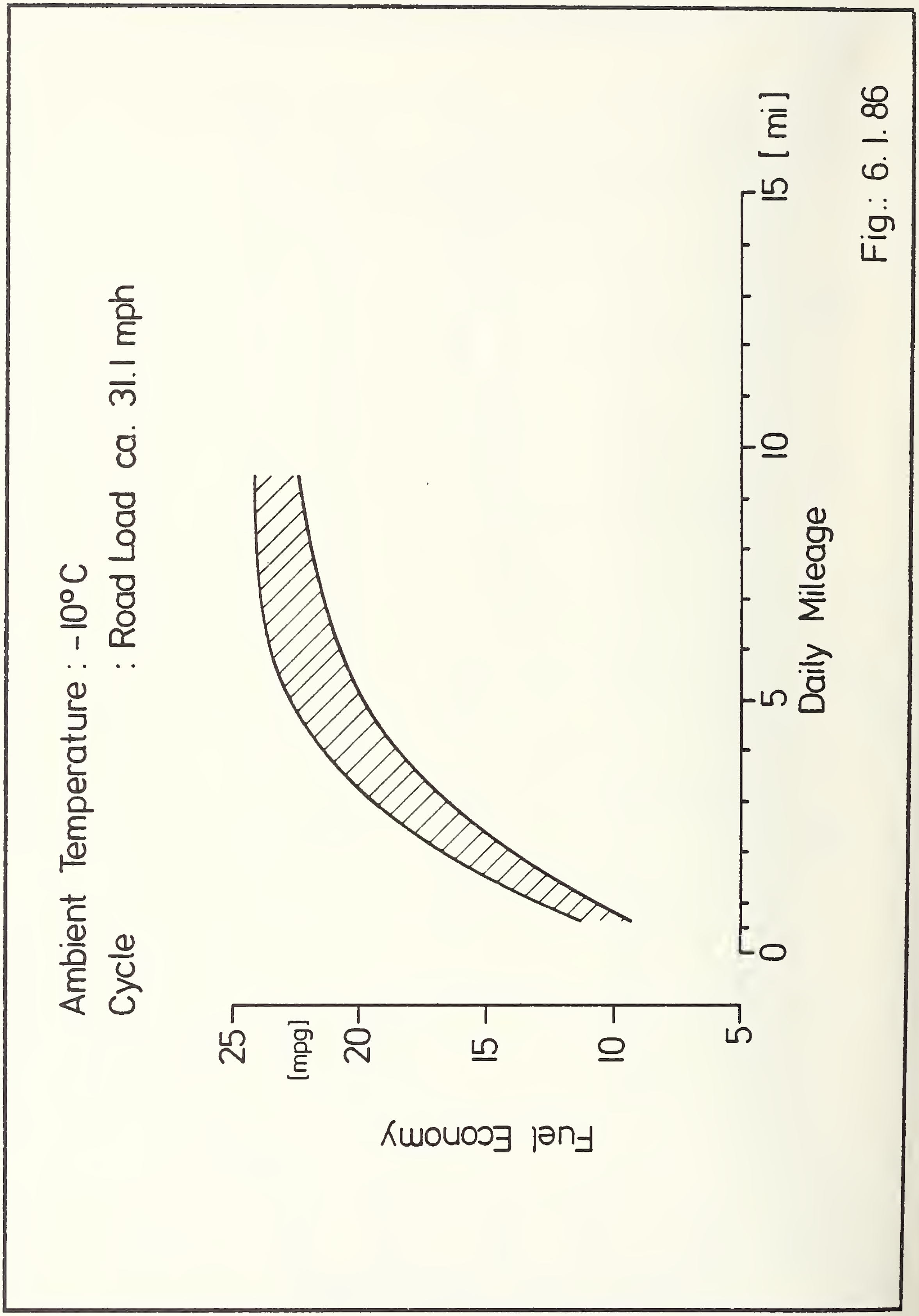

Ambient Temperature : -10°C
Cycle : Road Load ca. 31.1 mph
Fuel Economy
[mpg]
25
20
15
10
5
Daily Mileage
0
5
10
15 [mi]

Fig.: 6.1.86

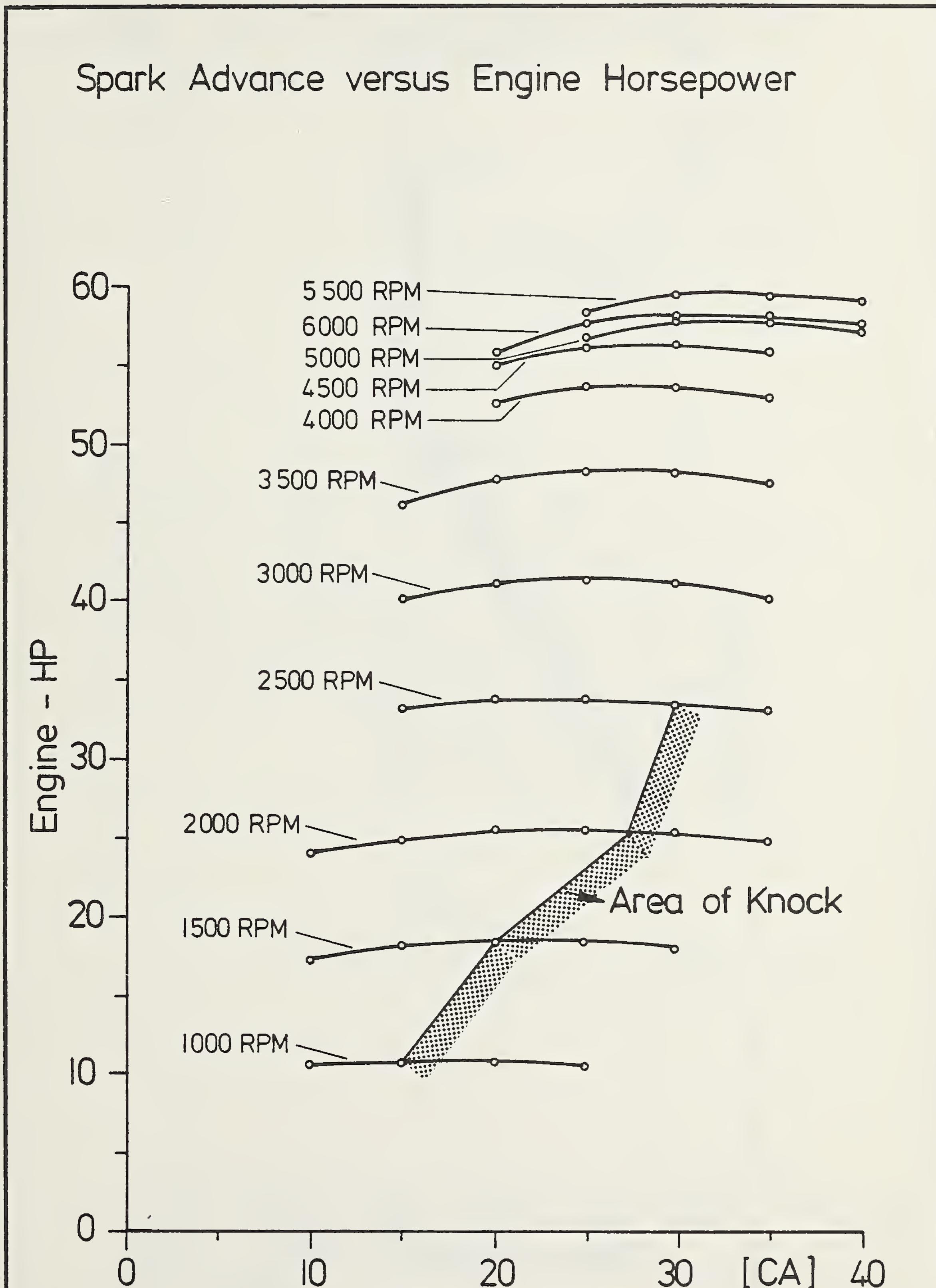

Spark Advance versus Engine Horsepower
5 500 RPM
6000 RPM
5000 RPM
4500 RPM
4000 RPM
3 500 RPM
3000 RPM
2500 RPM
2000 RPM
1500 RPM
1000 RPM
Area of Knock
Engine - HP
60
50
40
30
20
10
0
0
10
20
30
[CA]
40
Spark Advance

Fig.: 6. 1. 87

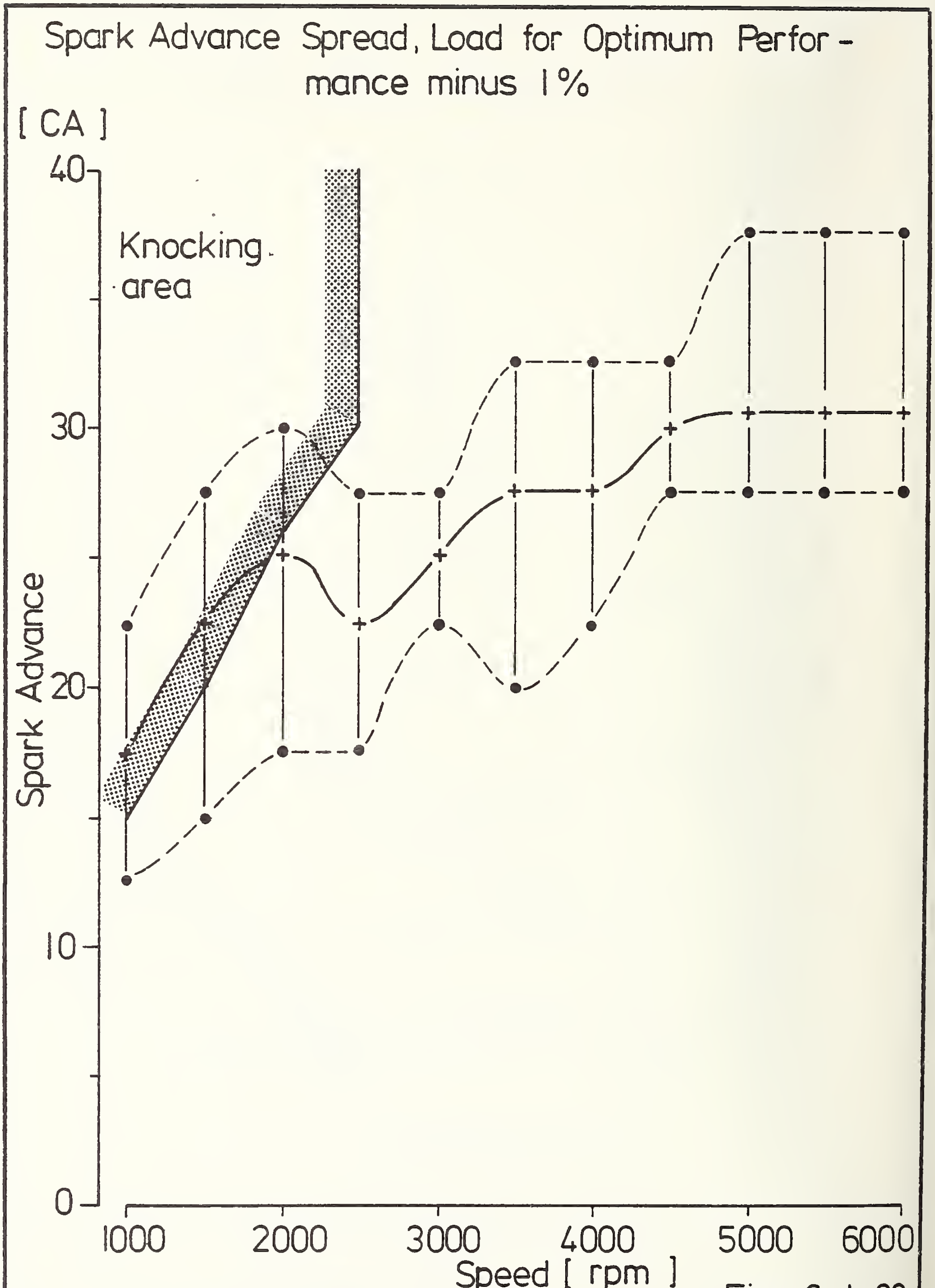

Spark Advance Spread, Load for Optimum Performance minus 1%
[CA]
Spark Advance
Knocking area
0
10
20
30
40
1000
2000
3000
4000
5000
6000
Speed [rpm]

Fig.: 6. 1. 88

Road Load Fuel Consumption versus Spark Advance

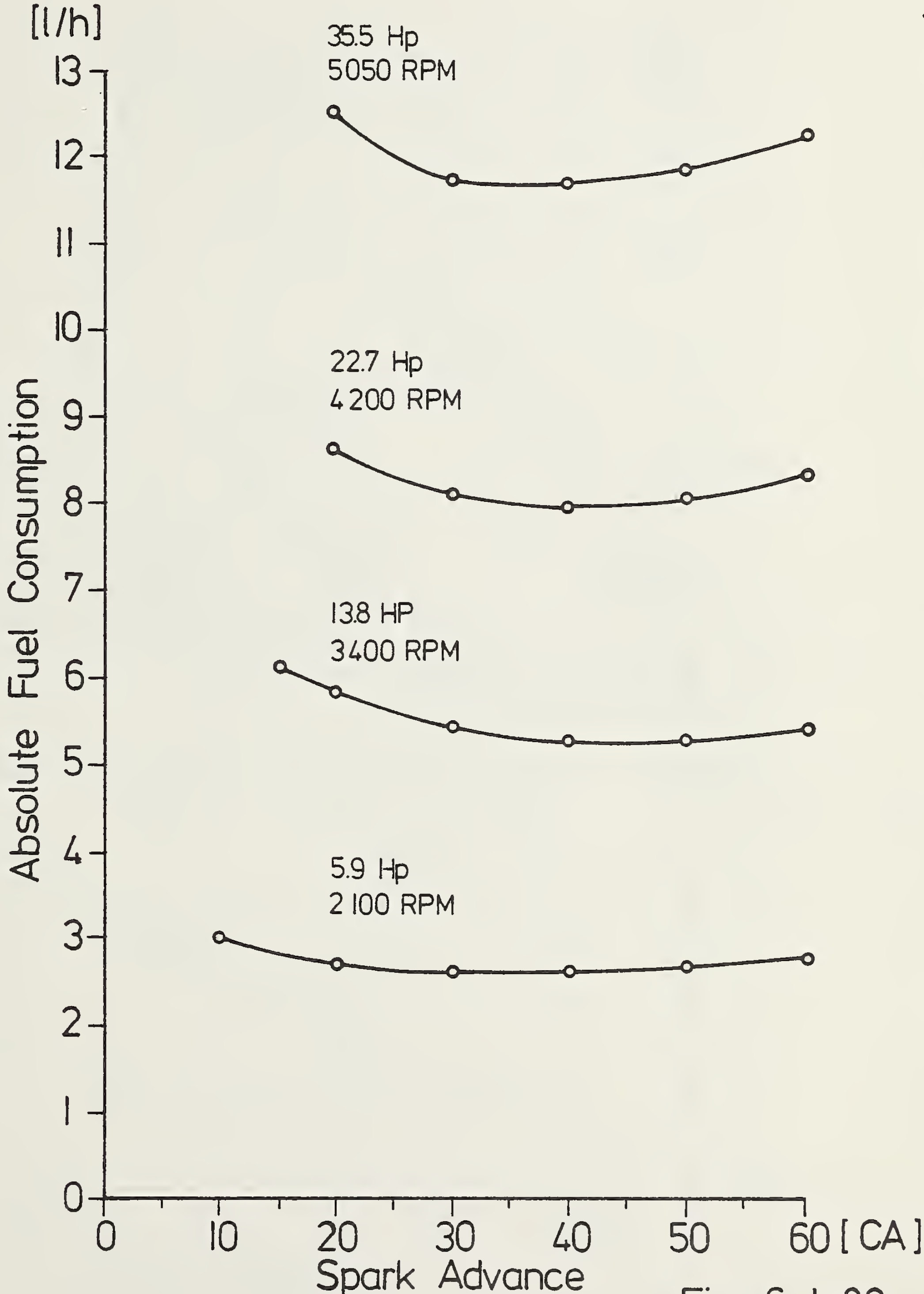
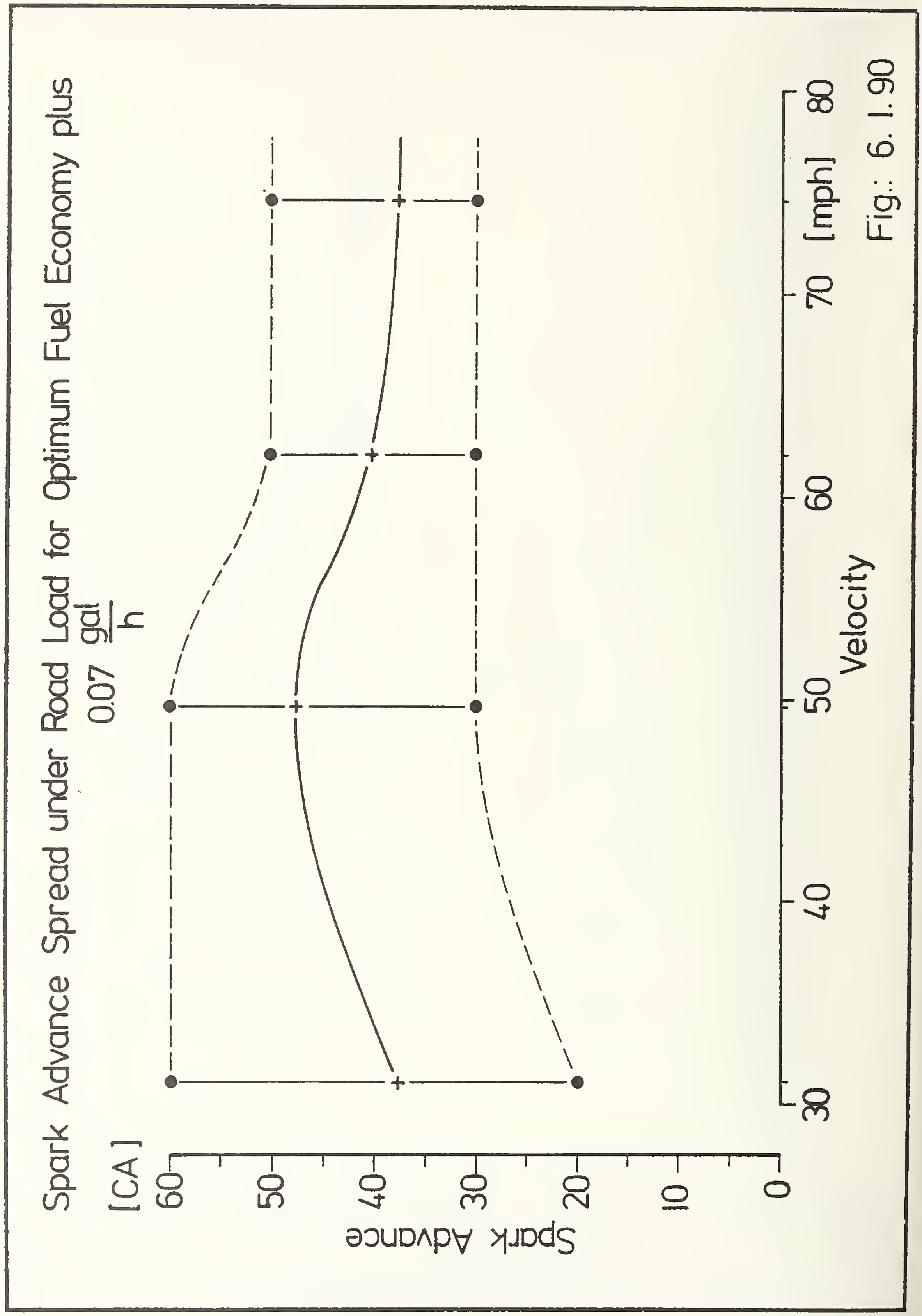


Fig.: 6.1.89



Spark Advance Spread under Road Load for Optimum Fuel Economy plus $0.07 \frac{gal}{h}$

Fig.: 6. 1. 90

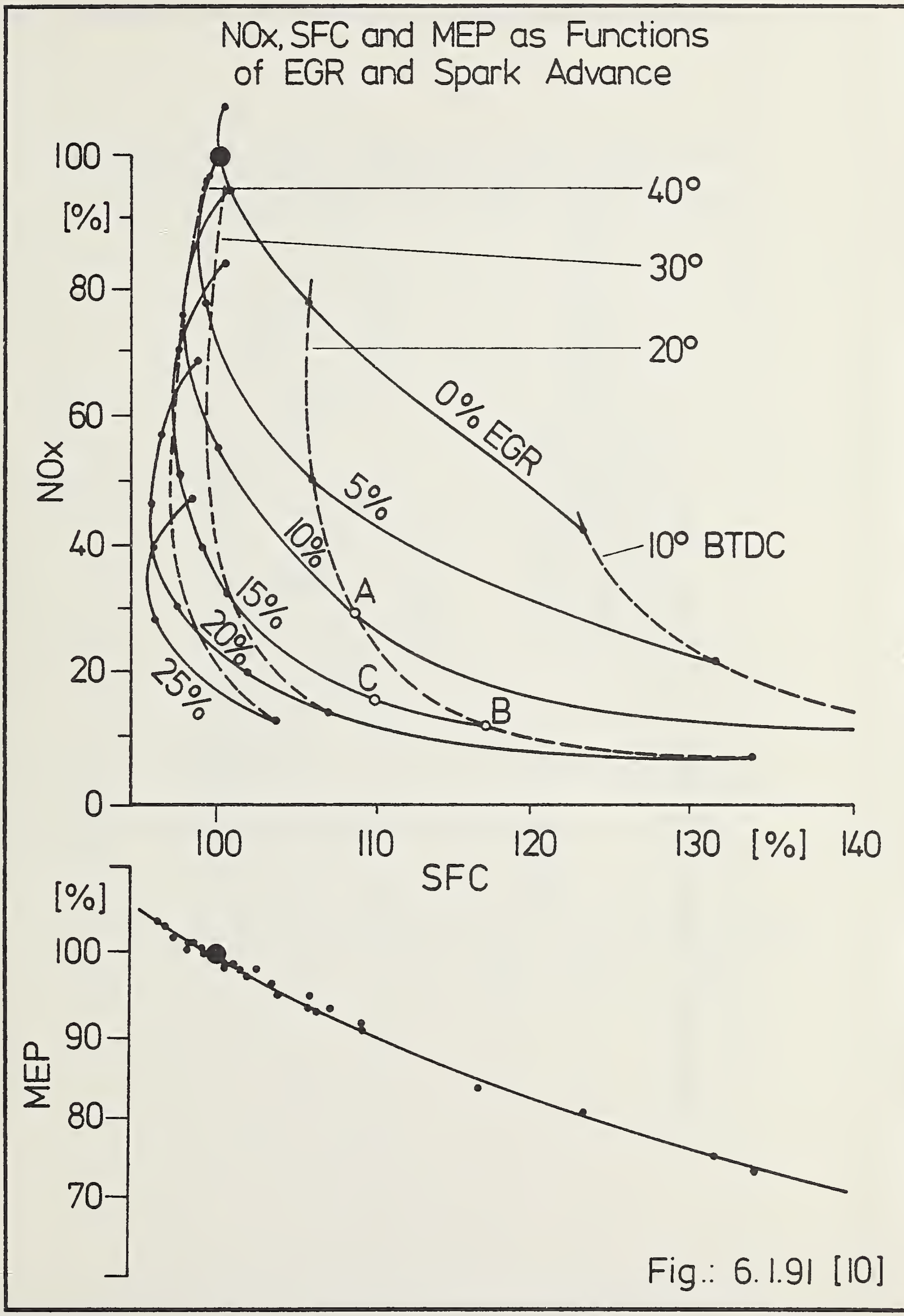

NOx, SFC and MEP as Functions of EGR and Spark Advance
100
[%]
80
60
40
20
0
NOx
40°
30°
20°
0% EGR
5%
10%
15%
20%
25%
10° BTDC
A
B
C
100
110
120
130
[%]
140
SFC
[%]
100
90
80
70
MEP

Fig.: 6.1.91 [10]

Estimated Interdependence of Fuel Flow, Air/Fuel Ratio, EGR, Spark Advance and Constant MEP

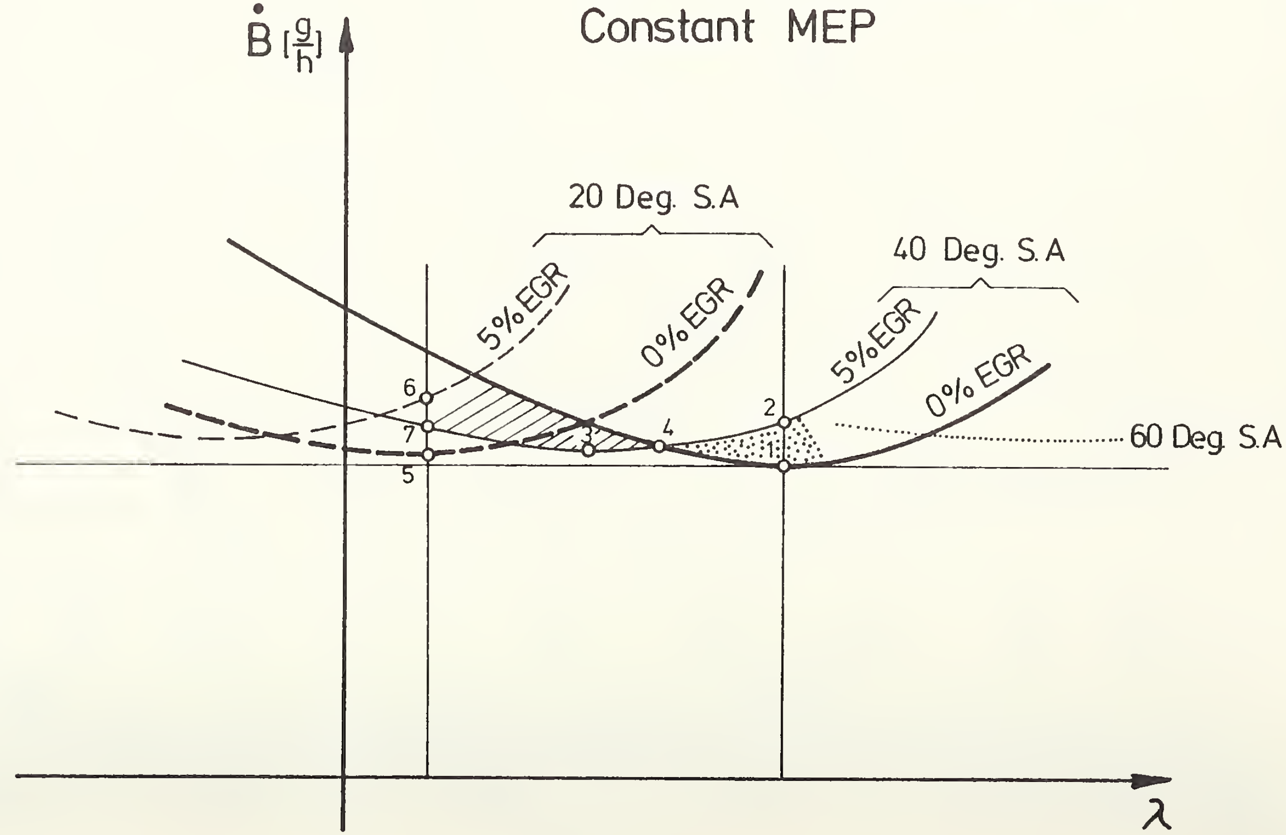


Fig.: 6. I. 92

Influence of Spark Advance and EGR on
Fuel Consumption and NOx Emissions

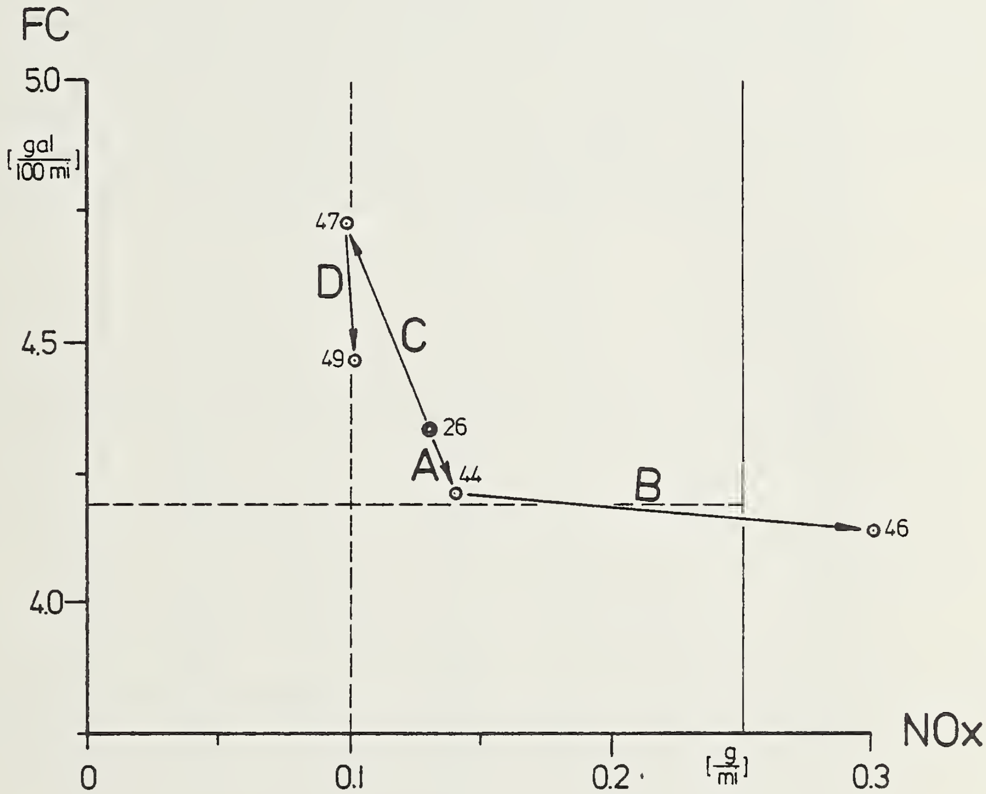


Fig.: 6. I. 93

Tab.: 6.1.1

Mod. Code	Type	License Plate	Displacem.	HP SAE net.	HP IW	Emission Tasks HC	Emission Tasks CO	Emission Tasks NOx	Inertia Weight 2250	Inertia Weight 3000	CVS-Emissions HC Test	HC AV.	CO Test	CO AV.	NOx Test	NOx AV.	Fuel Economy UDC Test	UDC AV.	UDC FC	UDC FC AV.	HDC Test	HDC AV.	Comb. Test	Comb. AV
01	Golf	WOB-VA 67	96.9	66.7	0.0296	uncontrolled			×		3.18		15.8		3.29		26.8		3.73		37.4		30.7	
01	"	"	"	"	"	"			×		3.21		21.7		3.67		24.1		4.15		33.9		27.7	
01	"	"	"	"	"	"			×		2.58	2.91	26.6	23.8	3.32	3.44	24.7	24.8	4.05	4.04	34.9	35.1	28.1	28.5
01	"	"	"	"	"	"			×		2.80		21.7		3.52		24.5		4.08		35.2		28.4	
01	"	"	"	"	"	"			×		2.76		33.3		3.41		23.8		4.20		34.3		27.6	
02	Golf	WOB-VA 67	96.9	66.7	0.0222	uncontrolled				×	2.84		24.1		4.39		23.2		4.31		30.3		25.9	
02	"	"	"	"	"	"				×	2.99		30.9		4.40		21.6		4.63		29.1		24.4	
02	"	"	"	"	"	"				×	3.14	3.10	28.7	29.3	4.34	4.34	22.7	22.5	4.41	4.45	29.0	29.8	25.2	25.3
02	"	"	"	"	"	"				×	3.39		25.8		4.33		22.9		4.37		29.5		25.4	
02	"	"	"	"	"	"				×	3.15		37.2		4.26		22.0		4.55		31.0		25.3	
03	Rabbit	WOB-VX 36	77.6	60.4	0.0268	uncontrolled			×		2.50		26.5		1.88		27.1		3.69		40.2		31.8	
03	"	"	"	"	"	"			×		2.60		29.5		1.84		27.0		3.70		41.8		32.1	
03	"	"	"	"	"	"			×		2.51	2.44	27.0	26.5	2.04	2.02	26.7	26.9	3.75	3.72	42.3	40.9	32.0	31.8
03	"	"	"	"	"	"			×		2.41		23.5		2.18		26.9		3.72		40.5		31.7	
03	"	"	"	"	"	"			×		2.18		26.0		2.18		26.9		3.72		39.8		31.5	
04	Audi 100	WOB-VR 93	96.9	84.9	0.0283	uncontrolled				×	3.75		24.5		3.65		21.4		4.67		32.9		25.3	
04	"	"	"	"	"	"				×	3.08		27.2		3.48		22.0		4.55		32.6		25.8	
04	"	"	"	"	"	"				×	3.17	3.13	30.8	27.8	3.94	3.74	20.0	21.1	5.00	4.74	30.1	31.3	23.6	24.8
04	"	"	"	"	"	"				×	3.09		27.2		3.88		21.0		4.76		30.6		24.4	
04	"	"	"	"	"	"				×	2.57		29.9		3.74		21.2		4.72		30.7		24.6	

Legend: AV. - Average ; FC - Fuel Consumption [gp100m]

Tab.: 6.1.2

Mod. Code	Type	License Plate	Displacem.	HP SAE net.	HP / IW	Emission Tasks HC	Emission Tasks CO	Emission Tasks NOx	Inertia Weight 2 250	Inertia Weight 3000	CVS-Emissions HC Test	HC AV.	CO Test	CO AV.	NOx Test	NOx AV.	Fuel Economy UDC Test	UDC AV.	UDC FC	UDC AV.	HDC Test	HDC AV.	Comb. Test	Comb. AV
05	Rabbit	WOB-VD 93	96.9	66.7	0.0296	1.5	15	3.1	×		0.54		8.54		2.19		24.5		4.08		34.1		28.1	
05	,,	,,	,,	,,	,,	,,	,,	,,	×		0.63		9.20		1.96		23.6		4.24		35.2		27.7	
05	,,	,,	,,	,,	,,	,,	,,	,,	×		0.57	0.59	5.29	7.26	2.01	2.18	24.9	23.9	4.02	4.18	36.8	35.1	29.1	27.9
05	,,	,,	,,	,,	,,	,,	,,	,,	×		0.58		6.60		2.40		23.3		4.29		34.3		27.2	
05	,,	,,	,,	,,	,,	,,	,,	,,	×		0.64		6.66		2.34		23.3		4.29		35.2		27.5	
06	Rabbit	WOB-VD 93	96.9	67.6	0.0225	1.5	15	3.1		×	1.07		12.7		1.88		20.9		4.78		30.5		24.4	
06	,,	,,	,,	,,	,,	,,	,,	,,		×	0.60		7.88		2.12		22.7		4.41		30.5		25.7	
06	,,	,,	,,	,,	,,	,,	,,	,,		×	0.60	0.71	8.72	9.69	1.95	2.00	22.1	21.7	4.52	4.61	30.8	30.5	25.3	25.0
06	,,	,,	,,	,,	,,	,,	,,	,,		×	0.65		8.80		1.97		21.4		4.67		30.5		24.7	
06	,,	,,	,,	,,	,,	,,	,,	,,		×	0.63		10.4		2.06		21.4		4.67		30.4		24.7	
07	Rabbit	WOB-VX 36	77.6	60.4	0.0268	1.5	15	3.1	×		0.75		6.79		1.50		24.4		4.10		35.0		28.3	
07	,,	,,	,,	,,	,,	,,	,,	,,	×		0.93		7.44		2.31		24.0		4.17		31.4		26.9	
07	,,	,,	,,	,,	,,	,,	,,	,,	×		0.97	0.95	8.76	8.08	2.40	2.32	24.7	24.4	4.05	4.12	34.8	34.0	28.4	27.9
07	,,	,,	,,	,,	,,	,,	,,	,,	×		1.31		10.1		2.52		24.4		4.10		34.6		28.1	
07	,,	,,	,,	,,	,,	,,	,,	,,	×		0.80		7.38		2.86		24.3		4.12		34.3		28.0	
08	Audi 100	WOB-VR 93	96.9	84.3	0.0281	1.5	15	3.1		×	1.17		10.5		1.85		25.3		3.95		30.0		27.2	
08	,,	,,	,,	,,	,,	,,	,,	,,		×	1.37		10.0		1.72		20.3		4.93		29.1		23.5	
08	,,	,,	,,	,,	,,	,,	,,	,,		×	1.43	1.26	11.3	10.6	1.79	1.77	21.2	21.9	4.72	4.59	29.7	31.6	25.4	25.6
08	,,	,,	,,	,,	,,	,,	,,	,,		×	1.15		10.7		1.80		20.0		5.00		35.2		24.8	
08	,,	,,	,,	,,	,,	,,	,,	,,		×	1.17		10.3		1.67		22.9		4.37		34.2		26.9	

Legend: AV. - Average ; FC - Fuel Consumption [gp100m]

Tab.: 6.1.3

Mod. Code	Type	License Plate	Dicplacem.	HP SAE net.	HP / IW	Emission Tasks			Inertia Weight		CVS-Emissions						Fuel Economy							
						HC	CO	NOx	2250	3000	HC Test	HC AV.	CO Test	CO AV.	NOx Test	NOx AV.	UDC Test	UDC AV.	UDC FC	UDC AV.	HDC Test	HDC AV.	Comb. Test	Comb. AV
09	Rabbit	WOB-VD 93	96.9	67.6	0.0300	0.9	9	2.0	×		0.38		3.57		1.54		23.2		4.31		31.9		26.5	
09	"	"	"	"	"	"	"	"	×		0.32		3.90		1.35		23.2		4.31		31.8		26.4	
09	"	"	"	"	"	"	"	"	×		0.37	0.34	5.38	4.03	1.19	1.40	23.6	23.3	4.24	4.30	33.0	32.3	27.1	26.6
09	"	"	"	"	"	"	"	"	×		0.32		3.95		1.37		22.9		4.37		32.5		26.4	
09	"	"	"	"	"	"	"	"	×		0.32		3.35		1.53		23.5		4.26		32.4		26.8	
10	Rabbit	WOB-VD 93	96.9	67.6	0.0225	0.9	9	2.0		×	0.46		6.92		1.82		22.0		4.55		31.2		25.4	
10	"	"	"	"	"	"	"	"		×	0.35		5.95		1.76		21.8		4.59		29.4		24.7	
10	"	"	"	"	"	"	"	"		×	0.43	0.40	7.90	6.50	1.75	1.76	21.4	22.0	4.67	4.54	30.1	30.3	24.6	25.1
10	"	"	"	"	"	"	"	"		×	0.39		6.86		1.52		22.6		4.42		30.3		25.5	
10	"	"	"	"	"	"	"	"		×	0.38		4.85		1.94		22.3		4.48		30.7		25.4	
11	Rabbit	WOB-VX 36	77.6	55.8	0.0248	0.9	9	2.0	×		0.66		5.91		1.93		25.0		4.00		35.7		28.9	
11	"	"	"	"	"	"	"	"	×		0.80		7.70		1.97		24.2		4.13		36.0		28.4	
11	"	"	"	"	"	"	"	"	×		0.63	0.72	6.25	6.67	1.20	1.52	24.7	24.6	4.29	4.12	35.8	35.5	27.3	28.3
11	"	"	"	"	"	"	"	"	×		0.79		7.51		1.21		23.6		4.24		34.1		27.4	
11	"	"	"	"	"	"	"	"	×		0.71		6.00		1.28		25.3		3.95		36.1		29.1	
12	Audi 100	WOB-VR 93	96.9	88.5	0.0281	0.9	9	2.0		×	0.62		7.94		1.74		20.3		4.93		29.9		23.7	
12	"	"	"	"	"	"	"	"		×	0.58		6.12		1.80		21.0		4.76		31.2		24.6	
12	"	"	"	"	"	"	"	"		×	0.63	0.62	8.89	7.63	1.72	1.74	21.2	20.9	4.72	4.79	31.3	30.8	24.8	24.4
12	"	"	"	"	"	"	"	"		×	0.63		7.98		1.70		20.9		4.78		31.1		24.5	
12	"	"	"	"	"	"	"	"		×	0.65		7.21		1.74		21.1		4.74		30.4		24.4	

Legend: AV. - Average ; FC - Fuel Consumption [gp100m]

Tab.: 6.1.4

Mod. Code	Type	License Plate	Diсplacem.	HP SAE net.	HP IW	Emission Tasks			Inertia Weight		CVS-Emissions						Fuel Economy							
						HC	CO	NOx	2 250	3 000	HC		CO		NOx		UDC				HDC		Comb.	
											Test	AV.	Test	AV.	Test	AV.	Test	AV.	FC	AV.	Test	AV.	Test	AV
13	Rabbit	WOB-VD 50	96.9	73.8	0.0328	0.41	3.4	1.0	×		0.15		0.62		0.21		25.4		3.94		34.0		28.8	
13	"	"	"	"	"	"	"	"	×		0.12		1.00		0.18		25.6		3.91		33.9		28.8	
13	"	"	"	"	"	"	"	"	×		0.11	0.12	0.64	0.81	0.19	0.18	25.2	25.3	3.97	3.95	33.6	33.7	28.4	28.5
13	"	"	"	"	"	"	"	"	×		0.12		0.85		0.21		25.2		3.97		33.7		28.4	
13	"	"	"	"	"	"	"	"	×		0.09		0.94		0.10		25.0		4.00		33.2		28.1	
14	Rabbit	WOB-VD 50	96.9	73.8	0.0246	0.41	3.4	1.0		×	0.14		1.23		0.15		23.2		4.31		30.8		26.1	
14	"	"	"	"	"	"	"	"		×	0.19		1.22		0.19		23.9		4.18		30.7		26.5	
14	"	"	"	"	"	"	"	"		×	0.13	0.16	0.98	1.24	0.17	0.18	23.5	23.5	4.26	4.26	30.6	30.6	26.2	26.2
14	"	"	"	"	"	"	"	"		×	0.18		1.06		0.17		23.3		4.29		30.5		26.1	
14	"	"	"	"	"	"	"	"		×	0.14		1.70		0.20		23.4		4.27		30.5		26.1	
15	Scirocco	WOB-VS 35	77.6	58.9	0.0262	0.41	3.4	1.0	×		0.14		1.57		0.17		26.5		3.77		34.9		29.7	
15	"	"	"	"	"	"	"	"	×		0.16		1.58		0.13		26.6		3.77		34.8		29.7	
15	"	"	"	"	"	"	"	"	×		0.14	0.15	1.29	1.52	0.13	0.13	26.7	26.6	3.74	3.68	35.0	34.8	29.9	29.7
15	"	"	"	"	"	"	"	"	×		0.17		2.00		0.11		26.4		3.39		34.7		29.5	
15	"	"	"	"	"	"	"	"	×		0.12		1.16		0.12		26.8		3.73		34.5		29.8	
16	Audi 5000	BS-JV 667	130.8	101.7	0.0337	0.41	3.4	1.0		×	0.09		1.44		0.18		16.3		6.14		24.2		19.1	
16	"	"	"	"	"	"	"	"		×	0.10		1.35		0.19		16.5		6.06		24.5		19.3	
16	"	"	"	"	"	"	"	"		×	0.13	0.10	1.25	1.30	0.18	0.18	16.4	16.3	6.10	6.14	23.9	24.2	19.1	19.1
16	"	"	"	"	"	"	"	"		×	0.06		1.19		0.19		16.1		6.21		24.2		18.9	
16	"	"	"	"	"	"	"	"		×	0.10		1.25		0.18		16.2		6.17		24.1		19.0	

Legend: AV. - Average ; FC - Fuel Consumption [gp100m]

Tab.: 6.1.5

Mod. Code	Type	License Plate	Dicplacem.	HP SAE net.	HP IW	Emission Tasks HC	Emission Tasks CO	Emission Tasks NOx	Inertia Weight 2250	Inertia Weight 3000	CVS-Emissions HC Test	HC AV.	CO Test	CO AV.	NOx Test	NOx AV.	Fuel Economy UDC Test	UDC AV.	UDC FC	UDC AV.	HDC Test	HDC AV.	Comb. Test	Comb. AV
17	Rabbit	WOB-VD 50	96.9	72.5	0.0322	0.41	3.4	0.4	×		0.16		1.09		0.11		24.1		4.15		32.4		27.2	
17	''	''	''	''	''	''	''	''	×		0.13		0.82		0.12		23.8		4.20		32.6		27.1	
17	''	''	''	''	''	''	''	''	×		0.12	0.13	0.95	0.87	0.13	0.12	24.0	24.0	4.17	4.16	32.4	32.4	27.2	27.2
17	''	''	''	''	''	''	''	''	×		0.11		0.74		0.11		24.1		4.15		32.5		27.3	
17	''	''	''	''	''	''	''	''	×		0.14		0.75		0.13		24.1		4.15		32.3		27.2	
18	Rabbit	WOB-VD 50	96.9	74.1	0.0247	0.41	3.4	0.4		×	0.22		1.91		0.10		20.4		4.90		30.5		24.0	
18	''	''	''	''	''	''	''	''		×	0.22		1.09		0.11		20.4		4.90		30.7		24.0	
18	''	''	''	''	''	''	''	''		×	0.13	0.16	1.68	1.51	0.14	0.13	20.7	20.5	4.83	4.89	28.4	29.1	23.6	23.6
18	''	''	''	''	''	''	''	''		×	0.10		1.40		0.14		20.4		4.90		28.4		23.4	
18	''	''	''	''	''	''	''	''		×	0.11		1.46		0.15		20.4		4.90		27.5		23.1	
19	Scirocco	WOB-VS 35	77.6	55.8	0.0261	0.41	3.4	0.4	×		0.26		1.57		0.09		25.7		3.89		39.1		30.4	
19	''	''	''	''	''	''	''	''	×		0.22		1.47		0.11		24.5		4.08		36.3		28.7	
19	''	''	''	''	''	''	''	''	×		0.27	0.23	1.84	1.66	0.10	0.10	24.3	24.6	4.12	4.07	35.2	36.5	28.2	28.8
19	''	''	''	''	''	''	''	''	×		0.21		1.65		0.11		23.7		4.22		35.8		28.0	
19	''	''	''	''	''	''	''	''	×		0.21		1.78		0.10		24.9		4.02		35.9		28.9	
20	Audi5000	BS-JV 667	130.8	102.6	0.0342	0.41	3.4	0.4		×	0.05		1.17		0.14		15.6		6.41		23.2		18.3	
20	''	''	''	''	''	''	''	''		×	0.09		1.58		0.14		15.4		6.50		22.8		18.0	
20	''	''	''	''	''	''	''	''		×	0.05	0.06	1.23	1.28	0.15	0.15	15.5	15.5	6.50	6.44	23.0	23.0	18.1	18.2
20	''	''	''	''	''	''	''	''		×	0.05		1.26		0.15		15.6		6.41		23.0		18.2	
20	''	''	''	''	''	''	''	''		×	0.05		1.17		0.15		15.6		6.41		23.1		18.3	

Legend: AV. - Average ; FC - Fuel Consumption [gp100m]

Modification Code	Test [mpg]	AV. [mpg]	Modification Code	Test [mpg]	AV. [mpg]	Modification Code	Test [mpg]	AV. [mpg	Modification Code	Test [mpg]	AV. [mpg]	Test
01	24.6		05	25.5		09	24.2		17	27.2		UDC
01	24.7		05	25.5		09	25.0		17	26.5		
01	26.6	25.3	05	24.7	25.2	09	25.0	24.7	17	24.9	26.5	
01	26.0		05	25.8		09	25.1		17	26.3		
01	24.5		05	24.8		09	24.0		17	27.6		
01	34.4		05	35.5		09	34.2		17	35.9		SET*
01	34.7		05	35.9		09	34.5		17	36.5		
01	34.3	34.6	05	36.2	36.2	09	34.5	34.4	17	35.6	35.7	
01	34.4		05	36.6		09	34.6		17	34.8		
01	34.2		05	36.6		09	34.0		17	35.7		
01	38.6		05	41.4		09	39.7		17	41.2		HDC
01	39.2		05	41.6		09	39.8		17	42.1		
01	38.8	38.8	05	41.6	41.8	09	40.3	39.8	17	41.5	41.3	
01	39.3		05	42.1		09	39.9		17	40.3		
01	38.2		05	42.2		09	39.2		17	41.2		

* Sulphate Emission Test
AV - Average

Tab.: 6.1.6

Variation of Inertia Weight

Engine/Vehicle	Drivetrain	IW [lbs]	Fuel Economy [mpg] UDC	UDC Hot Start	UDC Projection Basted on Engine Map	HDC	HDC Projection Basted on Engine Map
Mod. Code 03	D (see Tab.: 5.1.2)	2000	—	30.3	29.7	36.0	35.9
"	D	2 250	27.2	29.2	28.7	35.3	34.8
"	D	2500	—	—	27.5	—	33.8
"	D	2 750	—	—	26.4	—	32.8
"	D	3000	24.5	26.6	26.0	32.2	31.9

Tab.: 6. 1. 7

Engine Speed [rpm]	1.3 l ; 59 HP						1.6 l ; 74 HP						2.2 l ; 102 HP					
	throttle open			throttle closed			throttle open			throttle closed			throttle open			throttle closed		
	Power required for Engine Motoring [kW]	Full Load Power [kW]	Motoring Power / Full Load Power [%]	Power required for Engine Motoring [kW]	Full Load Power [kW]	Motoring Power / Full Load Power [%]	Power required for Engine Motoring [kW]	Full Load Power [kW]	Motoring Power / Full Load Power [%]	Power required for Engine Motoring [kW]	Full Load Power [kW]	Motoring Power / Full Load Power [%]	Power required for Engine Motoring [kW]	Full Load Power [kW]	Motoring Power / Full Load Power [%]	Power required for Engine Motoring [kW]	Full Load Power [kW]	Motoring Power / Full Load Power [%]
1000	0.9	7.3	12.3	1.3	7.3	17.9	3.5	8.8	40.0	4.5	8.8	51.4	3.8	13.5	28.3	3.5	13.5	26.0
1400	1.3	9.9	13.1	2.1	9.9	21.1	4.4	12.1	36.5	5.2	12.1	43.1	4.8	19.3	24.8	4.4	19.3	22.7
1800	2.3	14.1	16.3	3.0	14.1	21.3	5.0	17.2	29.0	6.4	17.2	37.1	6.0	24.1	24.9	5.7	24.1	236
2200	3.2	18.3	17.5	4.0	18.3	21.9	6.0	22.1	27.2	7.5	22.1	34.0	7.7	31.7	24.3	7.7	31.7	24.3
2600	4.2	22.1	19.0	5.4	22.1	24.5	7.3	26.8	27.3	9.0	26.8	33.6	9.8	37.8	25.9	9.5	37.8	25.1
3000	5.4	26.4	20.4	6.5	26.4	24.6	8.8	32.3	27.3	10.6	32.3	32.8	12.3	44.8	27.4	12.0	44.8	26.8
3400	6.7	30.5	22.0	8.0	30.5	26.2	10.6	36.3	29.2	12.2	36.3	33.6	15.0	51.2	29.3	14.7	51.2	28.7
3800	8.2	33.1	24.8	9.3	33.1	28.1	12.7	40.6	31.2	13.8	40.6	34.0	18.2	58.1	31.3	16.8	58.1	28.9
4200	10.0	35.3	28.4	10.8	35.3	30.6	15.0	44.6	33.7	15.7	44.6	35.2	22.0	65.7	33.5	19.8	65.7	30.2
4600	12.1	38.2	31.7	12.6	38.2	33.0	16.8	47.6	35.4	17.6	47.6	37.0	26.4	70.1	37.7	23.0	70.1	32.8
5000	14.7	40.8	36.0	14.3	40.8	35.1	17.6	51.4	34.3	19.8	51.4	38.6	30.8	72.7	42.4	26.4	72.7	36.3
5400	16.8	42.3	39.8	16.4	42.3	38.8	23.5	54.1	43.4	22.2	54.1	41.0	35.3	74.8	47.2	30.0	74.8	40.1
5600							25.8	54.1	47.7	24.2	54.1	44.7						
5800	19.5	43.1	45.2	18.3	43.1	42.5							40.2	72.9	55.2	33.7	72.9	46.2
6000	21.8	43.3	50.3	19.3	43.3	44.6							43.3	69.8	62.0	35.8	69.8	51.3

Friction and Ventilation Loss vs. Speed

Tab.: 6.1.8

Variation of Axle Ratio

Engine / Vehicle	Transmission Ratios 1. through 4. Gear	Axle Ratio	Fuel Economy [mpg] UDC Hot Start	UDC Projection Based on Engine Map	HDC	HDC Projection Based on Engine Map
Mod.Code 03	D (see Tab.: 5.1.2)	3.27	31.2	30.8	37.4	37.6
"	D	3.48	—	30.1	—	36.6
"	D	3.70	—	29.4	—	35.7
"	D	3.90	29.2	28.7	35.3	34.8
"	D	4.11	28.4	—	33.5	—

Tab.: 6.1.9

Engine / Vehicle	Transmission Ratios Gears					Axle Ratio	Remark	Fuel Economy [mpg]	
	1.	2.	3.	4.	5.			UDC	HDC
Mod. Code 03	3.46	1.94	1.29	0.75	—	3.90	4 Gears 1.÷3. Standard 4. Gear as Overdrive	29.1	38.2
"	"	"	"	0.97	—	"	4 Gears Standard	28.7	34.8
"	"	"	1.37	1.10	0.91	"	5 Gears Sports	28.2	35.4
"	"	"	1.23	"	"	"	5 Gears Sports Variation in 3. Gear	28.9	35.5
"	"	"	1.37	"	0.75	"	5 Gears Sports 5. Gear as Overdrive	28.6	38.2
"	"	"	1.23	0.97	"	3.27	5 Gears Economical	28.9	35.5

Drivetrain Variations

Projections Based on Engine Map

Tab.: 6. 1. 10

TABLE OF ELECTRICAL POWER CONSUMPTION (RABBIT; MANUAL TRANSMISSION)

(Amps at 13 volts as rated)

CONSUMER			"POSSIBLE CURRENT"
1.	Lowbean		8.4 A
2.	Parking Lights Tail Lights Side-marker Lamp Turn Signal System Stop Lamp Instrument Lighting License Plate	 (3.3 A x 0.1 *) (3.3 A x 0.1 *)	4.1 A
3.	Ignition Coil Automatic Choke		3.4 A
4.	Windshield Wiper	- Position I 2.8 - Position II 4.0	2.8 A
5.	Rear Window Heating		9.6 A
6.	Radiator Fan	(7.7 A x 0.5 *)	3.9 A
7.	Radiator Fan	(with air conditioner)	13.8 A
8.	Air Blower	- Position I 2.2 - Position II 3.9	2.2 A
9.	Air Conditioner (with magnetic clutch)	- Position I 12.7 A - Position II 17.5 A - Position III 27.5 A - Position IV 32.7 A	17.5 A
	Consumption at full load and 13 Volts		
	Rabbit without Air Conditioner		34.4 A
	Rabbit with Air Conditioner		61.8 A

* Relative Cyclic Duration Factor < 1

Tab.: 6. I. II

No.	Power Steering System	Delivery Volume per Rotation [cm^3]	Controlled Constant Flow [$\frac{l}{min}$]	Max. Pressure [bar]	Energy - Requirement of Pump [W]	Energy Percentage of I	Fuel Economy Loss [$\frac{gal}{100\ mi}$]	Remark
I	Vane-Type Pump, Open-Centered Control Valve	13.5	6	75	690	100	0.0030	US - Standard
II	Vane-Type Pump, Open-Centered Control Valve	8.5	6	75	419	61	0.0017	Audi 5000 Serial
III	Radial-Piston Pump, Open-Centered Control Valve	4.3	3,5	150	349	50	0.0015	
IV	Radial-Piston Pump, Closed-Centered Control Valve	2.7	—	140 ÷ 180	284	41	0.0012	Well Known Technology
V	Radial-Piston Pump Closed-Centered Control Valve without Leakage, Controlled Constant Pressure	2.7	—	140 ÷ 180	97	14	0.0004	Advanced Technology

Tab.: 6.1.12

COMBUSTION CHAMBER DATA

	Bowl in Piston	Cup-Shaped Piston	Wedge Combustion Chamber and Flat Piston (Standard)
Surface-to-Volume Ratio (cm^{-1})	2.97	3.09	3.15
Percentage of Quench Area Relative to Piston Area	-	46.5	20.2
Height of Quench Zone (mm)	-	1	1.5
Compression Ratio	8.69:1	8.39:1	7.88:1

Tab.: 6.1.13

Valve Timing Variations

Test No.	Camshaft No	Same Valve Timing as	Exhaust Valve Opens BTD	Exhaust Valve Closes TDC	Intake Valve Opens TDC	Intake Valve Closes BTD	Cam Overlap Deg	Cam Spread Deg	Cruise Fuel Economy [mpg] 31,1 mph	62,1 mph	74,5 mph
I	1	IX	18°b	14°b	2°b**	2°a*	-12	97.5	49.6	32.7	25.7
II	2	V; XI	2°b	2°a	2°b	2°a	4	90	50.1	32.2	25.0
III	3	XII	14°a	18°a	2°b	2°a	20	82.5	47.1	31.2	——
IV	3	XIII	2°b	2°a	18°b	14°b	20	82.5	49.6	32.2	24.6
V	2	II; XI	2°b	2°a	2°b	2°a	4	90	50.1	32.0	25.6
VI	1	VIII	2°b	2°a	14°a	18°a	-12	97.5	49.6	31.8	25.3
VII	1		13°a	17°a	29°a	33°a	-12	97.5	49.6	——	——
VIII	1	VI	2°b	2°a	14°a	18°a	-12	97.5	48.5	——	——
IX	1	I	18°b	14°b	2°b	2°a	-12	97.5	50.6	——	——
X	2		13°a	17°a	13°a	17°a	4	90	49.6	——	——
XI	2	II; V	2°b	2°a	2°b	2°a	4	90	50.1	——	——
XII	3	III	14°a	18°a	2°b	2°a	20	82.5	48.0	——	——
XIII	3	IV	2°b	2°a	18°b	14°b	20	82.5	49.6	——	——
XIV	3		17°b	13°b	33°b	29°b	20	82.5	50.6	——	——

* after
** before

Tab.: 6. 1. 14

FE Comparison with and without Acceleration Pump

Engine	Drivetrain		Inertia Weight	Acceleration Pump	Fuel Economy [mpg] UDC Hot Start	HDC
Mod. Code 03	D	(see Tab.: 5.1.2)	2 250	with	29.2	35.3
"	D		3 000	"	26.6	32.2
"	D		2 250	without	28.5	34.6
"	D		3 000	"	26.0	31.8

Tab.: 6.1.15

6.2 EMISSIONS

6.2.1 Methodology

There are two methods being used by VW by means of which the emissions of engine/vehicle systems are determined. The first method is the one described in the Federal Register, which involves testing complete vehicles on dynamometers according to the 1975 Federal Test Procedure.

The second method of determining the emissions of an engine/vehicle system involves testing the engine on an engine dynamometer controlled by the VW Programmed Control System (VW PCS). After having been brought to a temperature of +20°C, the engine is started cold, run through an Urban Driving Cycle (UDC) of 1372 seconds and after 10 minutes run through another 1372 seconds UDC, hot start this time.

Of course, the PCS must be fed beforehand with the data of the vehicle, especially the final drive ratio, the transmission ratios, the dynamic diameter of the tires, and finally the drag coefficient, the cross-sectional area, and the inertia weight.

While the engine is running, measurements controlled by the PCS are taken of the concentration of pollutants in the exhaust gas as well as of the quantity of air and fuel used. From those data, the emissions are computed. Thus, two emission ratings are computed from the UDC cold and from the UDC hot.

The method of computing from these recordings emission results which correspond to those from the 3-bag tests run on a chassis dynamometer is the same mathematically and physically valid method which was already described in Chapter 6.1.1 in connection with our fuel economy computations. Here again, the emission figures obtained on an engine dynamometer and on a vehicle dynamometer are bound to be the same. Results obtained from engine dynamometer tests offer an extra advantage in that they are more easily reproduced, because on an engine dynamometer there are none of the individualistic influences exerted by human drivers on vehicle dynamometers (see also Chapter 3.2.3). Therefore, readings taken on an engine dynamometer are much less erratic.

To illustrate this point, we compared in Fig. 6.2.1 tests which were performed on two engines whose concept and design were identical but whose tuning differed slightly (see Table 6.2.1). It is easy to see that engine dynamometer tests are better reproducible.

As work on this Contract progressed, engines were subjected to engine dynamometer tests whenever there was some development work to be done, which usually happened with the low exhaust emission concepts. From the time when work on this Contract began to this very day, VW does and did not manufacture any engines capable of meeting exhaust emission standards as low as these.

However, we already had the requisite knowhow for research engines and we had to bring the engines down to the levels required. This can be done much faster on an engine dynamometer than on a vehicle dynamometer, because all parts on which exhaust emissions depend are easily accessible, for, as can be seen from Fig. 6.1.1 through 6.1.7 of Chapter 6.1.1, results can be analyzed step by step, and cold starts are possible at twohour intervals, as both lubricant and coolant can be cooled down artificially. It is not necessary, therefore, to wait out the entire stabilization period of 12 hours normally required by an engine installed in a vehicle.

The third method, the projection of fuel economies and emissions from steady-state engine maps described in Chapter 6.1.1, is according to our experience, practically worthless as far as computing emissions is concerned. Emissions are largely dependent on instationary processes which are generally not taken into account in projections of this kind. Therefore, although this method is quite valuable in computing fuel economies it is not used in practice for computing emissions and determining the factors which influence them.

For evaluating the sulfate emissions of our vehicles the 1975 Federal Test Procedure was used again. In addition to this, our test program included HDC's as well as the Sulfate Emission test (SET) suggested by EPA for this specific purpose.

To achieve identical test conditions the vehicle are preconditioned. Unleaded fuel, complying with Federal Register specifications and containing less than 0,1 weight % sulfur, had been used. In these tests, sulfates and total sulfur emissions were measured, together with the controlled exhaust emissions.

In order to determine the sulfate emissions the exhaust gas is diluted in a dilution tunnel and filtered. The sulfate content in the particulate filter is determined analytically.

Figures 6.2.2 and 6.2.3 show the dilution tunnel which also permits measurements of engines of larger displacements. This tunnel is designed in accordance with the EPA system and consists of several parts to facilitate cleaning. The overall length of the tunnel is six meters; its interior diameter is 0,45 m. There is an aperture in the tunnel cross section near the exhaust intake designed to provide turbulence and to mix exhaust gas and air. The probe for drawing isokinetic samples is located at the end of the tunnel.

Water-soluble sulfates are removed from the particulate filter by means of a water/alcohol solution.

The method of analysis involves adding barium chloranilate to sulfuric acid. This process produces chloranile acid of a high color intensity which is measured photometrically. The white deposit of barium sulfate must be filtered off before measuring.

The automatic measuring method suggested by the EPA is highly sensitive. Figure 6.2.4 shows the flow diagram of the analyzing equipment which uses liquid chromatography. The mobile phase (60 % isopropanol and 40 % water) is pressed continiously through a cation exchanger and barium chloranylate column by a high-pressure pump. Substrate absorption is measured with an UV-detector at 310 nm and recorded by a recording device. A six-way valve is used to feed the test solution into the columns through a sampling loop. The light absorption caused by the coloration is recorded with the recording device and serves as a measure of sulfate concentration. The method is calibrated by applying known quantities of sulfuric acid.

We related the results of our sulfate emission measurements to the emission control concepts in connection with which they were made.

The EPA program for determining HCN emissions comprises UDC, SET, and HDC tests as well as 30 minutes of normal idling at 925 $\pm$ 75 rpm, and 30 minutes of idling at high speed (2000 rpm). All measurements must be taken with the engine adjusted in the normal way (certification tuning) and with the engine tuned to the rich side (malfunction).

The scope of this program is extraordinarily large; it should be condensed slightly. We have found that the tests run with the engine idling do not furnish any significant information; the same applies to the malfunction tests. Although the HCN emissions recorded during the malfunction tests are indeed higher than those found with the engine tuned to certification requirements, they still are so far below the EPA limit (Garage Criterion) that we left this aspect entirely out of consideration, limiting the scope of our HCN tests to the exact sequence of driving cycles prescribed by EPA for their sulfate emission test.

HCN test samples are drawn from the dilution tunnel (see Fig. 6.2.3). After dilution, part of the exhaust gas flow is diverted and conducted through two washing bottles filled with soda lye. The amount of HCN absorbed in these bottles is then analyzed by the pyridine-pyrazolone method: A random part of the absorbent solution is mixed first with a buffer solution and with a T-cholramine solution, then with the pyridine-pyrazolone reagent. The liquid then assumes a blue color, the intensity of which is measured photometrically at 620 nm. The method is calibrated by applying standardized potassium cyanide solutions.

To determine the amount of noise emitted by the vehicles we measured both the exterior and the internal noise levels. The exterior noise level was measured at idle, at a constant-speed drive-by (30 mph), and at an acceleration drive-by, with the vehicle accelerating from 30 mph at a distance of 25 ft (7.6 m) from the microphone. Our tests were run according to SAE J 986 A, with the microphone at a distance of 50 ft (15.25 m), the noise level being recorded in dB (A) units of sound pressure. Fig. 6.2.5 shows a sample record.

When measuring the internal noise level we applied the same method, the only difference being that the microphone was located in the place of the driver's head.

For the exterior noise level measurements at idle, the microphone was located 19.7 in (0.5 m) from the center of the radiator grille.

The exterior and internal noise level measurements at idle were performed once per modification code, whereas the exterior and internal noise level measurements at a cruising speed of 30 mph and at acceleration driveby from 30 mph were repeated 10 times.

From the drive-by records, we extracted only the peak average noise level for further use. Copies of all records as well as lists of the internal and idle noise level measurements are to be found in the Appendix.

Any vibration occurring in reciprocating piston engines are mainly caused by unbalanced mass forces and moments. For the measurement of vibrations the entire power plant is suspended from springs whose natural frequencies ($\geqslant$3 cps) are well below any of the frequencies to be analyzed. This is done to eliminate any possibility of suspension system reactions interfering with the vibration behaviour of the power plant itself (Fig. 6.2.6).

Under its own power, the engine is run from idle to maximum speed with piezoelectric acceleration transducers registering the vibrations. Synchronous filters pick out the interesting orders (in four-cylinder engines, for instance, second-order vibrations only), recording vibration accelerations over engine speed. It is even possible to record vibration amplitudes provided that the proper kind of diagram is used.

Fig. 6.2.7 shows an example of such a recording taken from the 1,3 l engine. All records are included in the Appendix. On the horizontal axis of the diagram, we have the crankshaft speed together with its corresponding second-order frequency; the vertical axis on the left shows the acceleration, whereas the one on the right shows the amplitudes. We can see from this diagram that the amplitude registered at the point of measurement remains throughout at a nearly constant 8×10^{-2} mm, and the angle of phase is nearly 22°. Assuming that we want to establish a relationship between the phase and the upper dead center of a certain piston the reference signal to be used for that purpose would be the position of this piston in the firing sequence. However, all angle-of-phase measurements have to be rectified by adding the angle between the upper dead center and the ignition point. This angle changes with the engine speed and the load.

6.2.2 Methane Content Correction Factor

There is no dispute about methane being a non-reactive, harmless hydrocarbon constituent found in the exhaust gases of motor vehicles. For this reason, all steps taken by the automotive industry to meet the strict standards of the future are taken on the assumption that it is permissible to subtract the methane content from the total hydrocarbon emission and that the 1980 HC limit of 0.41 gpm does not include the total amount of methane.

This opinion is corroborated by a number of statements made by EPA in the course of the past few years, and by the fact that it is general practice in California, where a HC standard of 0,41 gpm is already in force. For this reason, all future considerations should be grounded on this assumption.

There is general agreement that the vehicles constituting the baseline from which the statutory standard of a total of 0,41 gpm HC (THC) was computed had in their exhaust a CH_4 content of about 5 %. Therefore, the equivalent non-methane HC standard (NMHC) should be approximately 0,39 gpm.

The great significant of the Methane Content Correction Factor (MCCF) is demonstrated by a study involving the Beetle convertible which was undertaken by VW on behalf of the California Air Resources Board. The technical description of this vehicle is given in Table 6.2.2. Having clocked mileages between 10,000 and 13,500 miles in factory traffic, on the open road, and in city traffic, and having been serviced regularly, ten of those vehicles were subjected to CVS 75 exhaust emission tests, while the methane content was measured by gas chromatography. The test results, which are average figures from two separate measurements each are listed in Table 6.2.3.

Following the CARB memorandum of July 7, 1976, we computed from these figures that the

$$MCCF = 0.83.$$

Consequentely, the total hydrocarbon emissions of this engine family (including methane) may exceed the standard of 0.41 gpm, reaching as high as

$$0.39 : 0.83 = 0.47 \text{ gpm HC}$$

without emitting more pollutant HC than 0.39 gpm.

Differences of this size - 15 to 20 % - are of extraordinary importance if emission standards are low, because the eventual certification or rejection of an entire engine family may depend on it.

There are no MCCF data available concerning the vehicles which were investigated under this Contract. For this reason, the following section deals with the total hydrocarbon emissions only. This, however, should not obscure the fact that the future of these vehicles will also be vitally affected by the Methane Content Correction Factor.

6.2.3 Engineering Goals

In 1973, VW used a vehicle whose performance in a large number of emission tests had been found to be very level to carry out comparative tests on dynamometers installed in several different test laboratories, some EPA labs among them. The results are shown in Fig. 6.2.8.

$\bar{X}$ is a mean drawn from 13 tests made at the VW facilities in Wolfsburg, Germany. There is a 95 % probability of all further tests producing results between $+t_s$ and $-t_s$.

Later on, the vehicle was flown to the US, where it was subjected to a number of dynamometer tests in the order given below:

2 dynamometer tests at the Olson lab;
3 dynamometer tests at the EPA research lab;
3 dynamometer tests at the EPA certification lab;
4 dynamometer tests at VW's mobile lab;
2 dynamometer tests at the EPA research lab;
2 dynamometer tests at the EPA certification lab;
5 dynamometer tests at VW's Los Angeles lab, and
3 dynamometer tests at the CARB lab.

After this, the vehicle was brought back to Germany, where another 6 tests were run at VW's Wolfsburg facilities. The result of these tests appears at the right edge of the diagram as a mean value and confidence level. No changes were made on the vehicle over the entire period. The results of the retests performed by VW after the vehicle had been to the US show that its performance had indeed remained level.

While in the US, the results of some tests went beyond the confidence range:

EPA research lab (1 x HC);
EPA certification lab (1 x NO_x);
VW mobile lab (2 x CO);
VW Los Angeles lab (1 x NO_x and HC);
CARB lab (1 x NO_x).

In two cases, however, the kind of deviation found was such that a systematic error was probably involved. These two cases are:

1) All three HC measurements performed during the first series of tests at the EPA certification lab; and

2) The three HC and CO tests performed at the CARB lab.

This illustrates that it is by no means unneccessary to reckon with systematic errors when setting engineering goals, for the entire engine family would have failed the EPA as well as the CARB tests if any of the tests named above had been part of a certification test.

For this reason, VW had two engineering goals, one in which systematic errors are taken into account, and one in which they are not taken into account. The latter engineering goal is based on results, incorporating the dispersions which may occur even if laboratory vehicles only are concerned. No deterioration factors and no deviations occurring between laboratory vehicles and production vehicles are involved.

It is, of course, basic that systematic errors cannot be foreseen. Meanwhile, however, we have sufficient experience to define a margin delimiting their occurrence, but even if such an additional safety margin is incorporated this does not mean complete safety from systematic errors.

The engineering goals of VW for the model year 1975 pertaining to 49 States and California were:

$HC/CO/NO_x$	Emission Level	Engineering Goal Incl. Retest but No Syst. Error	Engineering Goal Incl. Retest and Syst. Error
49 States	1.5/15/3.1	1.2/12/2.6	1.0/10/2.1
California	0.9/9.0/2.0	0.7/7.0/1.7	0.6/6.0/1.4

After development work towards these engineering goals had been done, the emission test vehicles for the 1975 model year produced emissions as shown in Fig. 6.2.9 and 6.2.10. Fig. 6.2.9 indicates that although there was no systematic error between VW and EPA, the safety margin of HC was nearly exhausted. As the NO_x emissions recorded by EPA were outstandingly constant by normal standards, the mean of VW's NO_x emissions even slipped to the boundary of the confidence range of the mean of the EPA NO_x tests.

The results shown in Fig. 6.2.10 (California) are quite a lot worse. The CO tests of Type 1 as well as the NO_x tests of both Dasher and Rabbit go to the very limit of the safety margin, but above all there are obvious systematic errors occurring between the HC- and NO_x-measurements of VW and EPA. The proof of this is that VW's mean values are not within the confidence range of the EPA mean, and EPA's mean values are not within the confidence range of the VW mean. It was a sheer stroke of luck for VW that the results of the EPA tests deviated downwards from those of the VW tests. Had the reverse been the case, VW would have failed the Californian certification test.

This proves yet again that it is very useful to incorporate the effects of systematic errors in your engineering goals, this becoming all the more important as the standards to be met grow stricter.

In addition there are deviations between test vehicles in the development stage and production vehicles.

Results shown in Fig. 6.2.11 contain these deviations and the normal spread of emission test results. We obtained these results from 101 California production vehicles equipped with catalysts of the 1975 model year. The HC values of all vehicles were far below the HC standard. Even if the mean HC value were to increase from 0.26 to 0.55 the HC values of all vehicles would still remain below the HC standard, assuming that the deviation remains the same.

The mean CO value of 3 is not too low because the worst result was still below the standard. The same can be said for the mean NO_x value of 1.3, although one vehicle result slipped above the standard.

Taking the production deviations into consideration but ignoring the systematic error, the factors between standards and engineering goals would be

HC = 0.6
CO = 0.33
NO_x = 0.7

With the necessary margin for the systematic error mentioned above the engineering goals for 1975 standards without deterioration are as follows:

$HC/CO/NO_x$	Emission Standard	Engineering Goal Incl. Production Deviations and Retest, No Syst. Error and Deter. Factor	Engineering Goal Incl. Production Deviations, Retest and Syst. Error No Deter. Factor
49 States	1.5/15/3.1	0.9/5/2.2	0.75/4.2/1.8
California	0.9/9.0/2.0	0.54/3/1.4	0.46/2.6/1.2

In our work on this Contract we did not try to meet these Engineering Goals. In other words, we did not develop the concepts designed to meet the standards of '76 (Modification Codes 5 - 12) towards these Goals because these concepts were either production concepts or close to it, which means that they might well deviate from the original Engineering Goal but would have to meet the Standards at all costs.

With the advanced concepts (Modification Codes 13 - 20), on the other hand, it was our task to prove that they were capable of meeting the Engineering Goals in the development stage. To arrive at suitable Engineering Goals we had to take into account the following factors:

1) Compared to production vehicles, the amount of VW's experience concerning vehicles of this kind is small;

2) There is no experience concerning the deviation of test results obtained from vehicles of this kind at VW laboratories and elsewhere;

3) The deterioration factors of the carburation-ignition-combustion system are rather insecure, and

4) The deterioration factors pertaining to the oxygen sensor and to the 3-way catalyst are still largely unknown.

To begin with the Research Standard, it is obvious that a factor of 2.0 as regards HC, 3.0 as regards CO and 1.7 as regards NO_x between engineering goals and standards is necessary to compensate for influences 1) and 2).

Over and above that, there are the deterioration factors 3) and 4). Conclusions in this matter cannot be drawn from certification tests of former model years because:

a) We have some but no representative results from EPA-Durability Testing

b) EPA-Durability Testing is not comparable with field experiences.

Therefore, in order to get the deterioration factors of 3-way-catalysts and closed loop systems we have to make certain assumptions:

If a dual-bed catalytic system is used to meet the statutory standard, i.e. without a closed loop but with the very low fuel economy of such systems, it seems to be realistic to assume a deterioration factor for HC, CO and NO_x of 1.4.

If a closed-loop system is used, the deterioration factor of HC and CO should be the same. But our catalyst experts tell us that the low NO_x standard will cause an additional deterioration factor of no less than 2.0, which is mainly due to the fact that the NO_x conversion efficiency drops within the λ-window. From all these factors, our engineering goals are computed as follows:

Emission	Standard	Factor for Production and Measuring Deviations incl. Syst. Error	Assumed Deterioration Factor of 3-way-cat. and closed loop syst.	Resulting Engineering Goal
HC	0.41	2.0	1.4	0.15
CO	3.4	3.0	1.4	0.81
NO_x	0.4	1.7	2.4	0.1

However, engineering goals like this would be extreme. They are hardly to achieve.

Therefore, we took another way. As we are responsible for the performance of the entire system, we postulated the following objectives in order to arrive at an engineering goal:

a) In cooperation with all concerned, including the EPA, Measuring and Testing will have to ensure that even under low emission standards the dispersion of test results remains narrow enough so that a vehicle whose emissions amount to less than half the standard after 50,000 miles remains below the standard, no matter where or under what circumstances it is tested.

   Developing and Manufacturing have to ensure that each production vehicle of a model whose emissions in the development stage amount to less than half the standard after 50,000 miles durability test remains below the standard, no matter where or under what circumstances it is tested.

b) Research and Development will have to ensure that the deterioration factors of the carburation-ignition-combustion system do not exceed 1.

c) The suppliers of oxygen sensors and catalysts will have to ensure that the HC and CO deterioration factors measured over 50,000 miles do not exceed 1.

d) The suppliers of oxygen sensors and catalysts will have to ensure that the NO_x deterioration factor measured over 50,000 miles does not exceed 2.

These objectives are optimistic but not entirely unrealistic. We will have to see whether it is possible to attain them or not. Under the conditions stated above, the engineering goal pertaining to the Research Standard of 0.41/3.4/0.4 would have to be 0.2/1.7/0.1.

Basically, the '81 Federal Standard of 0.41/3.4/1.0 is subject to the same considerations as the Research Standard because they both require the same kind of technology, i.e. a 3-way catalyst and an oxygen sensor and closed loop plus a clean-up catalyst. Presuming that the objectives are the same as those given above, we can set the engineering goal for Measuring and Testing and Research and Development as well as for the suppliers of oxygen sensors and catalysts at 0.2/1.7/0.25. All statements made so far are based on test procedure requirements.

Variations of test results and deviations of production vehicles are physical properties. A possible way to eliminate these difficulties would be to ensure that the true emissions, i.e. the average emissions of the production vehicles and not of individual vehicles, pass muster.

6.2.4 Emissions of Uncontrolled and Current Engines

The concepts which produced the emissions listed in Table 6.2.4 are described in detail in Chapter 4, whereas the vehicle data are given in Chapter 5.

As we are producing vehicles equipped with fuel injection as well as carburetor engines, both meeting the '76 Standards, we were faced with the question which concept to use for our tests. We decided in favor of the carburetor concept, because on an average they are slightly closer to minimum fuel consumption. This is because for reasons of driveability carburetor engines cannot, as a rule, be tuned quite as lean as fuel injection engines, which are capable of being tuned so far to the lean side that they will meet the '76 Standards without any need for catalysts. If they do, however, their tuning is so lean that they move away from minimum fuel consumption. For this reason, carburetor concepts with their richer tuning are generally preferable from the fuel economy point of view.

Because of this relatively rich tuning there is not enough oxygen in the exhaust gas, which is why we had to instal a self-aspirating secondary air system to supply additional oxygen.

Catalysts of a diameter of 4 in. and a length of 3 in. are sufficient to meet the 1976 Federal Standards. These catalysts are those which are specified in the VW certification, their nobel metal content being no less than 0.75 g. The nobel metal consists of platinum and rhodium in a mass ratio of 12:1. The substrate is a ceramic monolith of 200 cells per sq. in.

The same catalyst was used to meet the '76 California Standards, the only difference being that its length was 6 in.

In order to meet the '76 Federal Standards, the vehicles of the 3,000 lbs IW class had to be fitted with deceleration controls to lower the HC emissions during coasting, and with on-off EGR to reduce the NO_x emissions. The deceleration control consisted of a simple closing damper, whereas the EGR system was controlled by a pneumatic on-off valve whose operation was governed by the cooling water temperature and the throttle angle.

To meet the HC standard of 0.9 and the NO_x standards of 2.0 stipulated by the '76 California Standard, all engine/vehicle combinations had to be equipped with closing dampers and on-off EGR valves.

Fig. 6.2.12 shows all emission test results in the order of engine modifications. In the controlled vehicles, a certain similarity in the CO and HC emission curves is noticeable, which is not evident in the Uncontrolled vehicles. One would generally expect the NO_x emissions to develop contrary to the CO emissions, but this expectation came to nothing as well. What we are facing here, therefore, are instances of genuine scatter, which are due to a plethora of different influences.

Fig. 6.2.13 shows the averages as well as the maximum dispersions obtained from the 5 tests run on each engine/vehicle combination. One tendency is clearly distinguishable: If only the inertia weight is changed of all the parameters of a vehicle, as is the case between Modifications 01 and 02, 05 and 06, and 09 and 10, the emissions increase in every case. The fact that under the '76 Federal Standard (1.5/15/3.1 HC/CO/NO_x) the NO_x emissions constitute an exception to this rule is easily explained, because in order to get the NO_x emissions of the 3,000 lbs vehicle at all below the Standard, EGR had to be used. This was not the case in the 2,250 lbs vehicle. The changeover from 2,250 to 3,000 lbs inertia weight entails a HC emission increase of between 7 and 20 %, whereas CO increases by 23 to 61 %, and NO_x by 26 %, always assuming, of course, that the data are comparable.

6.2.5 Emissions of Advanced Engines

The concepts which produced the emissions listed in Table 6.2.5 are described in detail in Chapter 4, whereas the vehicle data are given in Chapter 5.

As it seemed likely that a certain amount of development work would have to be done on these concepts, we decided in favor of fuel injection to ensure easy tuning and precise control. The fuel injection system used is a mechanical system made by Bosch (K-Jetronik).

Initially, we had hoped to meet the Engineering Goals of the '81 Federal and of the Research Standard with concepts mainly consisting of a fuel injection system and a 3-way catalyst and closed loop.

However, as our work progressed we found that this was possible only in the case of the small engine (Modifications 15 and 19) combined with the 2,250 lbs vehicle. The 1.6 l engine in combination with the 2,250 lbs vehicle already had to be fitted with a secondary air pump and a clean-up catalyst to meet the '81 Standard Engineering Goals, which meant that all other engine/vehicle systems had to be similarly equipped.

As fuel consumption is adversely affected by EGR as a general rule (see Chapter 6.1.13) we tried to make do without EGR as far as possible. In this, we were successful with the vehicles of the 2,250 lbs IW class, where we were able to meet both advanced standards. In the 3,000 lbs class, however, we had to use EGR to meet the NO_x Engineering Goals of both standards. We used a proportional EGR system as described in Chapter 4.

With the small engine we succeeded in meeting the '81 Standard without deceleration control. We feel sure that it would have been possible with the 5-cylinder engine (Modification Code 20) in view of its average HC emissions, too. However, we decided not to run any tests to verify this because we did not expect that the elimination of the deceleration control unit would improve fuel consumption in any way.

By using Modification Code 17 as a sample, we now intend to demonstrate the way in which our development work progressed towards the final emissions and fuel economy readings which are listed in Tables 6.1.4 and 6.1.5 as well as in Chapter 6.1.2. This will also serve to explain why some of the NO_x readings listed there are slightly in excess of the Engineering Goals. Quite simply, the reason is that adherence to the Engineering Goals would have led to a very drastic deterioration in fuel economy in some concepts, which is why we decided that we had better look for a compromise which was more promising as far as fuel consumption was concerned.

To begin with, a total of 18 tests was required to pinpoint the hardware described above and to approach our goal in cold-start tests. Test No. 18 was a cold-start test pure and simple (US '72 test), so that its results are comparable to those from the first and second bag of a CVS 75 test.

Following this, we ran a number of additional tests to approach the engineering Goal of 0.1 gpm NO_x (tests 18 through 26).

Table 6.2.6 shows the results of cold start tests 18, 19, 20, 21, 23, 24, 25, and 26. Test 22 was eliminated because a measuring instrument failed. Figure 6.2.14 shows the results of these tests in comparison to the parameter changes involved. Figure 6.2.15 represents a different kind of presentation.

To put this two Figures in perspective it is necessary to recall that tests 18 through 26 are cold start tests and were therefore performed according to the 1972 test procedure. To bring these results up to the procedural standard of 1975 they would have to be merged with the results of a corresponding number of hot start tests.

This process would involve multiplying the hot-start test results by a factor of 0.57, and the cold-start test results by a factor of 0.43. To get a proper picture of our 72 cold-start test results, we converted our 1975 engineering goals (0.2/1.7/0.1 or 0.25 of $HC/CO/NO_x$) to fit 1972 standards. We based this conversion on our experience that under these conditions the HC emissions recorded during a hot-start test will hardly be worse than 0.1 g/mile, and CO will not exceed 0.5 g/mile, and that there will be no difference at all between the cold-start and the hot-start NO_x emissions. Therefore, in accordance with the 1972 procedure our cold-start engineering goals are

$$\frac{0.2 - 0.57 \times 0.1}{0.43} = 0.33 \text{ (HC), and}$$

$$\frac{1.7 - 0.57 \times 0.5}{0.43} = 3.3 \text{ (CO),}$$

whereas NO_x = either 0.1 or 0.25.

These engineering goals have been entered in Figures 6.2.14 and 6.2.15. Our basic assumption was that both the HC and the CO goals represent absolute maximum limits, as does the NO_x engineering goal of 0.25. On the other hand, remaining below a NO_x emission of 0.1 g/mile is desirable in order to meet the Research Standard. We also aimed at remaining below the fuel consumption of Modification Code 01 (Uncontrolled) which in the UDC amounts to 24.8 mpg (see Chapter 6.1.2). Assuming that including the influence of a third sampling bag into the calculation brings a fuel economy improvement of about 0.9 mpg, an assumption which is amply corroborated by our experience, the fuel economy of Modification Code 01 calculated from the first and second bag is 23.9 mpg, which means that, in other figures, its fuel consumption is 4.18 gal/100 miles. These two figures have also been entered into Figures 6.2.14 and 6.2.15 and marked as Engineering Goals. The new Engineering Goals refer to cold starts. If the actual test results go beyond these goals, this indicates that three-bag tests may produce results which are even better than the original Engineering Goals. In other words: as all values were obtained from three-bag tests the average emissions should have been less than 0.2 g/mi in the case of HC, below 1.7 in the case of CO, and below 0.25 in the case of NO_x, whereas fuel consumption should have been better than 4.18 $\frac{\text{gal}}{\text{100 mi}}$.

Of the entire series of tests considered here, it was test number 18 which gave results below all cold-start engineering goals save that of 0.1 for NO_x. In this test, even the fuel economy was found to be superior to that of the uncontrolled vehicle.

We then took a number of steps specifically designed to reduce NO_x emissions without deterioration of the fuel economy of the engine. Throughout test number 18, approximately 150 l of secondary air were used per minute, injected in the beginning in front of, and 160 seconds later behind the TWC.

We now hoped to improve the NO_x emissions by injecting the secondary air behind the TWC right from the beginning of the cold start test. Although this did improve the NO_x emissions as expected, other emissions, especially those of CO, deteriorated so badly that they increased beyond the engineering goal. Therefore, this procedure was not acceptable. The fact that there was an improvement in fuel economy as well must be regarded as accidental and as nothing but an outcome of the normal dispersion of test results. In the following test (number 20) we injected secondary air before the TWC for no more than 40 seconds. For the same period, we deactivated the manifold-pressure controlled spark advance unit to get a higher exhaust gas temperature and therefore a faster catalyst warm-up. This naturally brought some improvements in the HC and CO emissions. There is hardly any deterioration of NO_x, and the fuel economy remains within the normal scatter band.

Test 21 brought a further improvement in the emissions of HC and CO because we injected secondary air in front of the TWC for 50 seconds while keeping the ignition timing advance unit deactivated. Although this constitutes a reason for an increase in the emissions of NO_x it does not explain the extent of that increase, which is most probably due to a slightly leaner mixture. After all, the fuel economy of test 21 is somewhat better than can be accounted for by the scatter bandwidth of the former test results. Test 23 brought an increase in fuel consumption and a corresponding increase in the emission of CO although we injected secondary air in front of the TWC for 60 seconds. That the deterioration of the emissions of hydrocarbons did not keep pace with the deterioration of CO emissions may be due to the fact that we deactivated the vacuum advance for 90 seconds starting with the first acceleration phase of the test.

In test 24, we deactivated the altitude sensor but left everything else as it was in test 23 because we thought that the altitude sensor affected the test result adversely. The outcome proved us to be right.

In test 25, we reduced the cold start fuel enrichment by increasing the control pressure for cold start at 20°C from 2.0 to 2.5 bar. We found that this brought about a marked deterioration in HC emissions during the cold start phase. This indicated combustion problems, which indeed existed to the extent that the fuel economy was not markedly improved, and that for acceleration the throttle had to be opened so far that the emissions of NO_x deteriorated.

For test 26 we brought the richness of the air/fuel mixture during the cold start phase back to its former level and adjusted the basic air/fuel ratio of the K-Jetronic fuel injection somewhat more towards the rich side by reducing the control pressure with the engine at operating temperature from 3.55 to 3.0 bar. As a result, NO_x emissions came down close to the lower engineering goal, the fuel economy is only slightly outside our engineering goal, and the emissions of both HC and CO are satisfactory.

In a number of isolated cold start tests we then strove to improve the results obtained from the engine adjusted as in test 26 by modifying the engine adjustment further. The main results of our efforts are shown in Table 6.2.7, Figures 6.2.15 and 6.2.17. Here again, our main goal was to improve fuel economy and NO_x emissions. Our considerations were band on the average derived from the results of 5 cold start tests run later on the engine adjusted as in test 26. This position is represented by the twice-encircled dot on Figure 6.2.17. The first important step in this investigation is test 44, which is run with the spark retard diaphragm deactivated. The HC emissions recorded in this test were close to the permissible limit because the first catalyst did not warm up properly; NO_x emissions deteriorated a bit, which can be due only to natural dispersion, whereas the fuel economy improved slightly, which was according to expectations. The next step (46) was to keep the spark retard diapragm deactivated while reducing the basic air/fuel ratio of the K-Jetronic injection system to a normal level of leanness by increasing the control pressure from 3.0 to 3.55 bar. This brought about a further improvement in fuel economy; CO and HC emissions were acceptable, but the emissions of NO_x rose to 0.3 g/mile, which exceeds the maximum permissible limit.

In the next test (47) we determined the influence of an EGR system in proportion to the engine air throughput (see Fig. 4.2.9) on the engine adjusted as in test 26. We found that in this way we were able to remain below the engineering goal of 0.1 g/mile NO_x, at the cost, however, of a fuel consumption which was higher by nearly 13 % than that of modification code 01, and by 10 % compared to the average of all tests run on the engine adjusted as in test 26. By advancing the ignition timing by 5°, which was done in the next test (49), we were again able to produce a marked improvement in fuel economy at the cost of a minimal deterioration in NO_x emissions. We then ran another test (58), which involved advancing the ignition timing by another 5° while keeping the quantity of EGR constant. As a result of this test we found not that further progress was possible in this way.

The hitherto optimum compromise between good fuel economy and good NO_x emissions seems to be the results of the two tests, 48 and 52, which involved an engine adjustment as in test 26, but with the ignition timing advanced by 5° and without any EGR. In this case, the fuel consumption is 1 % better than that produced by the uncontrolled engine, the only drawback being that the emissions of NO_x are inferior to the lower Engineering Goal.

For this reason, we selected the tuning of test No. 26 for use with Modification Code 17, because it is nearly exactly mid-way between the tuning of tests 48 and 52, which we think is an optimum compromise, and that of tests 49 and 58, where the NO_x Engineering Goal was met, but where the fuel economy goal was not approached closely enough.

Therefore we proceeded to run 5 cold-start, 5 hot-start, and 5 HDC tests on the engine adjusted as in test 26. The results of these tests are listed in Table 6.2.8.

Figure 6.2.18 is a presentation of the results of the cold start and Figure 6.2.19 of the hot-start tests, which are combined in Fig. 6.2.20 and 6.2.21.

The development work done to determine the optimum hardware and tuning of the other concepts ran more or less parallel to that of Modification Code 17. Table 6.2.9 shows what tuning was finally decided upon; Fig. 6.2.20 shows the emission results as a combination of cold and hot-start test results, following the order of modifications; and Fig. 6.2.21 shows the average values together with their maximum dispersions.

In the presentation of those results we used two different scales to facilitate comparing these figures with the emissions of the uncontrolled and current engines (Fig. 6.2.12 and 6.2.13). The individual tests shown that there is no correlation whatever between the three kinds of pollutant emissions. Scatter bandwidths seem to be entirely due to accident.

Here again, the rule applies that emissions increase with the inertia weight; this is illustrated by Modifications 13 and 14 as well as 17 and 18.

It is important that all averages are below the Engineering Goals of the '81 Standard (0.41/3.4/1.0) which are 0.2/1.7/0.25 HC/CO/NO_x (see Fig. 6.2.21). There was only one individual test in which CO was somewhat high (Table 6.2.5).

We were not quite as successful in our efforts to meet the Research Standard: One HC average (Mod. Code 19), all individual emission readings from which it was computed, and two more individual HC test results did not meet the Engineering Goal. There were three individual CO readings which were in excess of the engineering goal, two of which again were connected with Modification Code 19. It is, therefore, an inescapable conclusion that with a NO_x level as low as this (average 0.1 gpm), this engine, too, has reached a limit beyond which a clean-up catalyst has to be used.

As far as NO_x emissions are concerned, it was thought sufficient—for fuel economy reasons (see above, Fig. 6.2.17) merely to approximate the Engineering Goal of 0.1 gpm. This standard was fully met, however, by the average readings of Mod. Code 19. All other test results ranged from 0.1 to 0.15 gpm.

6.2.6 Sulfate Emissions

Exhaust emission tests performed on vehicles running on unleaded fuel which were equipped with oxydation catalysts has shown that the particulate emissions of these vehicles are comparatively high.

An analysis indicated the presence of sulfuric acid or sulfates, and it was concluded that the oxydation catalysts used for purifying the exhaust gas could be made to convert part of the sulfur contained in the fuel after combustion into sulfuric acid. This conclusion initiated a wide range of studies designed to determine the actual extent of sulfate emissions.

It was found that, generally speaking, sulfate emissions vary with the quantity of secondary air, that they have nothing to do with the space velocity in the catalyst, that there is no correlation between the emission of sulfates and the limited emissions, and that the ageing of a catalyst is accompanied by a reduction in sulfate emission, although the conversion rates of the controlled pollutants may still remain the same.

One of the most significant findings are that secondary air is an influential factor: An excess of secondary air means that O_2 is available for the conversion of SO_2 to SO_3 or SO_4. If, on the other hand, the amount of O_2 available barely suffieces for the conversion of HC and CO the SO_3 reaction will not take place. This is important as far as the practical application of TWC's is concerned, because this system does not allow for any excess O_2.

Although there can be no doubt that there are reaction mechanism involving the formation of sulfates which occur preferably at catalysts, the overall result of all sulfate emission studies performed by VW in the past, including those which form part of this Contract, is that there is no difference between the sulfate mass emissions of vehicles with and without catalysts. A general survey of all test results available furnishes the following mass emission ranges:

Vehicles without catalysts: 0.66 - 1.4 mgpm H_2SO_4,

Vehicles with Pt/Rh-catalysts: 0.31 - 1.5 mgpm H_2SO_4.

These figures do not vary with the driving cycle. The sulfur content of the fuels used varied between 0.024 and 0.03 per cent by weight.

According to our experience, the engine concept does not have any influence, either; in retests, the scatter bandwidth of the sulfate readings is so large that it is impossible to say whether the engine in question is fitted with a carburetor, a fuel injection system, an EGR system, etc.

Expressed as percentages, the SO_4 conversion rates measured in the various driving cycles compare as follows:
UDC : SET : HDC = approximately 0.7 : 1.0 : 1.2.
In spite of this, the mass emissions expressed in mgpm remain always more or less the same. This is because fuel consumption tends to improve as the average test speed increases.

All vehicles tested emit between 0.4 and 2 % of the sulfur contained in the fuel as SO_4; of the entire total of particulates emitted, SO_4 accounts for an average of 9 %.

The tests run under the terms of this Contract corroborate fully the results of our former tests. This time, we tested a number of Rabbit concepts equipped according to Modification Codes 01, 05, 09 and 17. The results are shown in Table 6.2.10, whereas the individual readings are entered in Fig. 6.2.22. Fig. 6.2.23 shows the average results and maximum dispersions obtained from the five tests run in each of the three cycles, UDC, SET, and HDC on each Modification Code. There are hardly any tendencies discernible at all; there is no relationship to the exhaust emission concept or the driving cycle. Very cautiously, the opinion might be advanced that catalyst concepts are more inclined towards low sulfate emissions than engines not equipped with catalysts, and that there might be a tendency for the sulfate emissions of engines equipped with 3-way catalysts (Mod. Code 17) to rise from UDC via SET to HDC. The latter phenomenen may be due to the fact that the average speeds of the UDC, SET, and HDC tests are 21, 35, and 48 mph, so that they, too, increase from UDC via SET to HDC, which means that the temperature of the catalyst increases as well.

Regarding the behaviour of the rate of conversion of SO_2 to SO_3 in a 3-way catalyst, we have two graphs furnished by DEGUSSA, Fig. 6.2.24 and 6.2.25. The first graph shows that with an air/fuel ratio of 1.0 in spite of a relatively low SO_3 formation rate there is still a tendency for the formation of SO_3, to speed up as the temperature increases. This is underlined by Fig. 6.2.25 which was recorded with a mixture somewhat leaner than $\lambda = 1$, and which indicates that there is an obvious relationship between the formation of SO_3 and the temperature of the catalyst.

Fig. 6.2.26 shows the SO_4 averages from Fig. 6.2.23 together with the average controlled emissions measured at the same time. There is no noticeable correlation between SO_4 and one of the controlled pollutants.

6.2.7 HCN Emissions

Engines tuned in a certain way and equipped with platinum-rhodium catalysts were found to emit more HCN than usual. This fact triggered of the inception of a test program designed to investigate the emission of HCN, its formation, its origin, and its relation to variations in the engine concept and tuning. Other uncontrolled exhaust gas components were to be investigated at the same time. In these tests, the concentration of HCN was determined essentially along the lines suggested by the Triangle Park EPA Research Institute, i.e. by pyridine-pyrazolone analysis (see Chapter 6.2.1).

In combination with a number of theoretical considerations, the results available serve to prove the existence of the following relationships:

a) HCN emissions decrease as the engine temperature increases;

b) If the engine is tuned to the rich side, HCN is formed at the catalyst;

c) If the engine is tuned to the lean side, the formation of HCN at the catalyst decreases;

d) The higher the speed of passage through the catalyst, the higher the emission of HCN will be.

e) In catalyst concepts with the engine tuned to the rich side, HCN emissions will increase until the air/fuel ratio is nearly stoichiometric. As soon as lambda reaches 0.95, it suddenly collapses down to nearly zero, where it remains while the engine runs on a lean mixture. In engines not equipped with catalysts, however, there is almost no connection between the emission of HCN and the air/fuel ratio (see Fig. 6.2.27 (11)).

Meanwhile, the EPA has set two standards limiting the HCN emissions of vehicles, the so-called "Freeway Criterion", which amounts to 300 mgpm, and the "Garage Criterion", which amounts to 15 mg per minute. The results of all HCN tests performed by VW so far are far below these criteria. The highest measurement ever recorded was 16 mgpm, or one twentieth of the Freeway Criterion, and 3 mg per minute, about one fifth of the Garage Criterion.

The tests run by us under the provisions of this Contract were limited to measuring HCN emissions in the UDC, SET, and HDC cycles, using the same vehicles as for the sulfate tests. The HCN and sulfate emission measurements were taken simultaneously. Table 6.2.11 shows the results of all individual measurements. Fig. 6.2.28 shows all individual readings, and Fig. 6.2.29 lists the averages and maximum dispersions obtained from each Modification Code and driving cycle. This last graph demonstrates with unusual clarity that HCN emissions drop as the catalyst temperature and the space velocity increase. This follows from the fact that, in the same engine modification, the HCN emissions in SET are lower than in UDC, and lower in HDC than in SET: Now, both the space velocity and the catalyst temperature increase with the average speed of a test, and the average speed of the UDC is 20 mph, the SET, 35 mph, and the HDC 48 mph. In addition to all this, the UDC is at a disadvantage compared to SET and HDC because it begins with a cold start and not with a hot start, as SET and HDC do.

The second important bit of information to be drawn from Fig. 6.2.29 is that HCN emissions will drop as the emission of controlled pollutants is reduced, and that they will not increase again after a closed loop is introduced. This is made even clearer in Fig. 6.2.30, where the HCN emissions are entered together with the emissions of the three controlled pollutants, and related to cycles and Modification Codes. HCN emissions will drop more or less parallel with CO, although the reduction at the point of transition from Uncontrolled engines to engines fitted with catalysts is not quite as drastic.

In view of the fact that the '76 Federal Standard (1.5/15/3.4 gpm $HC/CO/NO_x$, Modification Code 05) provides a relatively high incidence of operating modes involving mixtures noticeably richer than stoichiometric it is not to be marvelled at that some of the HCN emissions found under these circumstances are relatively high, which follows the trend set by Fig. 6.2.27.

The absolute maximum HCN emission recorded in the course of the studies performed under the terms of this contract again amounted to 16 mgpm, which is far below the Freeway Criterion, corresponding to 5 % of its limit.

Leaving this particular measurement out of consideration because it may have been due to a malfunction, we present the relationship between HCN and CO in Fig. 6.2.31, and it can be seen that the correlation holds true fairly persistently.

6.2.8 Noise

Table 6.2.12 lists the averages obtained from all noise tests. Fig. 6.2.32 shows the noise readings of the 1.6 l Rabbit, Fig. 6.2.33 those of the 1.3 l Rabbit, and Fig. 6.2.34 shows those of the 3,000 lbs vehicle. Each vehicle was tested at four different emission levels.

There are no noise measurements given of Modification Codes 02, 06, 10, 14 and 17 through 20, because the noise levels emitted by these concepts are perfectly identical with others which have been tested. The following concepts have the same noise levels:

01 and 02
05 and 06
09 and 10
13 and 14 + 17
15 and 19
16 and 20.

Copies of all noise level records and lists of all internal and idle noise measurements are to be found in the Appendix.

As Fig. 6.2.32 shows, the lowest noise emission readings belong to the vehicle whose emissions are lowest in other respects as well, the sole exception being the internal noise measured while accelerating from 30 mph.

Compared to the noise emitted by 01 and 13, the noise level of Modification Codes 05 and 09 increases as the vehicle is being run increasingly closer to idling speed. This is due to 05 and 09 being equipped with self-aspirating secondary air systems whose silencing had not yet been perfected when we installed them. They were responsible for some disagreeable noise, which became all the more noticeable as the flow of air became uneven, and pulsations of this kind are bound to become more intensive as the engine approaches idling speed. This is why, for instance, the internal noise measured at a cruising speed of 30 mph comes closer to that measured at idle that the noise level recorded while accelerating from 30 mph. There is, by the way, nothing remarkable about the noise generated by the self-aspirating secondary air system measured internally being not quite as intensive as that measured externally, because the engine and the passenger compartment are separated by soundproofing.

This phenomenen is not quite as well defined in the same vehicle equipped with the smaller engine (see Fig. 6.2.33). This may be because the total quantity of secondary air is smaller, so that the noise of the self-aspirating secondary air system was slightly damped. But even this graph shows that vehicles with low emissions also emit little noise, the only factor at variance being the exterior noise measured at idle.

From Fig. 6.2.34, which shows the noise emitted by the 3,000 lbs IW vehicle, we can see that Modification Codes 08 and 12 are also influenced by the self-aspirating secondary air system. However, the effect here is not as intensive as in the 1.6 l Rabbit because in these Modification Codes the secondary air system had already been effectively silenced.

The noise level of Modification Code 16 is further reduced because the bigger 5-cylinder engine is involved, and because the body of the vehicle is of very recent construction.

Figs. 6.2.35 through 6.2.38 present a survey of the average noise levels of all vehicles belonging to the same emission standards. Fig. 6.2.35, which is a comparison of all uncontrolled vehicles, indicates quite clearly that bigger engines and bigger vehicles lower noise levels.

In figs. 6.2.36 and 6.2.37, this impression is somewhat distorted because of the intensive noise emitted by the engines equipped with self-aspirating secondary air systems, whereas in Fig. 6.2.38 the discrepancy is due to the high internal noise level of Modification Code 13, which has already been discussed above and seems to be pertaining to one specific vehicle, because this vehicle, license plate WOB-VD 50, was tested for noise in Modification 13 only.

Generally speaking, larger vehicles will emit less noise, especially as far as the internal noise level is concerned, because it is quite natural that in a bigger vehicle care should be taken to ensure superior comfort.

This is underlined by a comparison between the low emissions of Modification Codes 04, 08, and 12 and those of 01, 05 and 09. The same engine was used basically in both cases, and in the heavier vehicle it had even been tuned to increase its specific power output.

6.2.9 Vibrations

The vibrations of engines are completely independent of the emission control concept, and largely independent of their torque and power output. Vibrations are related to design parameters and to the engine speed, which is why, under the terms of this Contract, it seemed sensible to measure only the vibrations of the baseline engines and not of the Modifications.

Unfortunately, we were not in a position to include the 5-cylinder engine in our report, so that we are limited to the 1.3 and 1.6 l 4-cylinder engines.

Basically, nine pick-up positions were provided for measuring the vibrations of the engines including clutch and transmission (see Fig. 6.2.39), the measurements being taken in the three basic axes, piston, crankcase and transversal.

The smaller engine has been measured for vibrations in the piston and transversal axes in positions 1 through 7 and 9, whereas the 1.6 l engine was measured in all three axes in positions 1, 2, 3, 6, 8, and 9. All test records, which are similar to Fig. 6.2.7 (see Chapter 6.2.1), are to be found in the Appendix.

From those records we extracted the acceleration figures (expressed in m/sec^2) pertaining to the engine speeds of 2,500 and 5,000 rpm. These figures are listed in Table 6.2.13. Some of those figures we used to outline the most significant trends in Figs. 6.2.40 through 6.2.46. From these figures, we can see that, by and large, the 1.6 l engine vibrates more intensively than the 1.3 l engine. This is due to a number of factors which are not a subject of this investigation. The decisive point is that bigger engines do not necessarily vibrate less intensively.

Figs. 6.2.40 through 6.2.43 show records of vibration accelerations measured versus the longitudinal axis of engine and transmission, the distance between the individual pick-up positions being approximately the same as that along the longitudinal axis of the engine.

We selected these particular pick-up positions because we wanted to present the piston-axis vibrations in positions 1, 5, 6, and 9, and the transversal vibrations in positions 2, 5, 6 and 9.

In line with our expectations we found that the alternator is subjected to a comparatively intense transversal acceleration, whereas generally speaking it is the piston axis along which the most powerful vibrations occur. As a rule, vibration acceleration increases with the engine speed. The fact that two lines in the 1.3 l graph are nearly identical shows that the accelerations recorded in pick-up positions 5 and 6 do not differ much, either along the piston or along the transversal axis, although position 5 is on the cylinder head on top of the engine, whereas No. 6 is located on the clutch casing on a level with the crankshaft.

Another general tendency is for the engines to vibrate less on their flywheel side than in front. This in mainly because much of the engine mass is concentrated in this area. Although in Fig. 6.2.42 there seems to be a belly in the vibration curve at position 6 (1.6 l 5000 rpm) this does not constitute an exception; it merely indicates that the piston axis vibrations measured at the alternator are of comparatively low intensity.

Due to the gyroscope effect of masses rotating especially as the generating forces are low, vibrations along the crankshaft axis are generally kept low (see Table 6.1.13). In spite of all this, the highest acceleration ever recorded by us did occur in the crankshaft axis, at the alternator of the 1.6 l engine. The explanation is that the mounting of the alternator on the engine block is to elastic.

In Figs. 6.2.44 through 6.2.46 we compared the vibrations of alternators, carburetors, and starters. Here again, the general rule applies in that the auxiliaries of the 1.6 l engine are exposed to more intensive vibrations than those of the 1.3 l engine. The main reason of the differences, however, is the rigidity of the individual suspension.

Reproducibility on Engine and Chassis Dynomometers

(Various Adjustments)

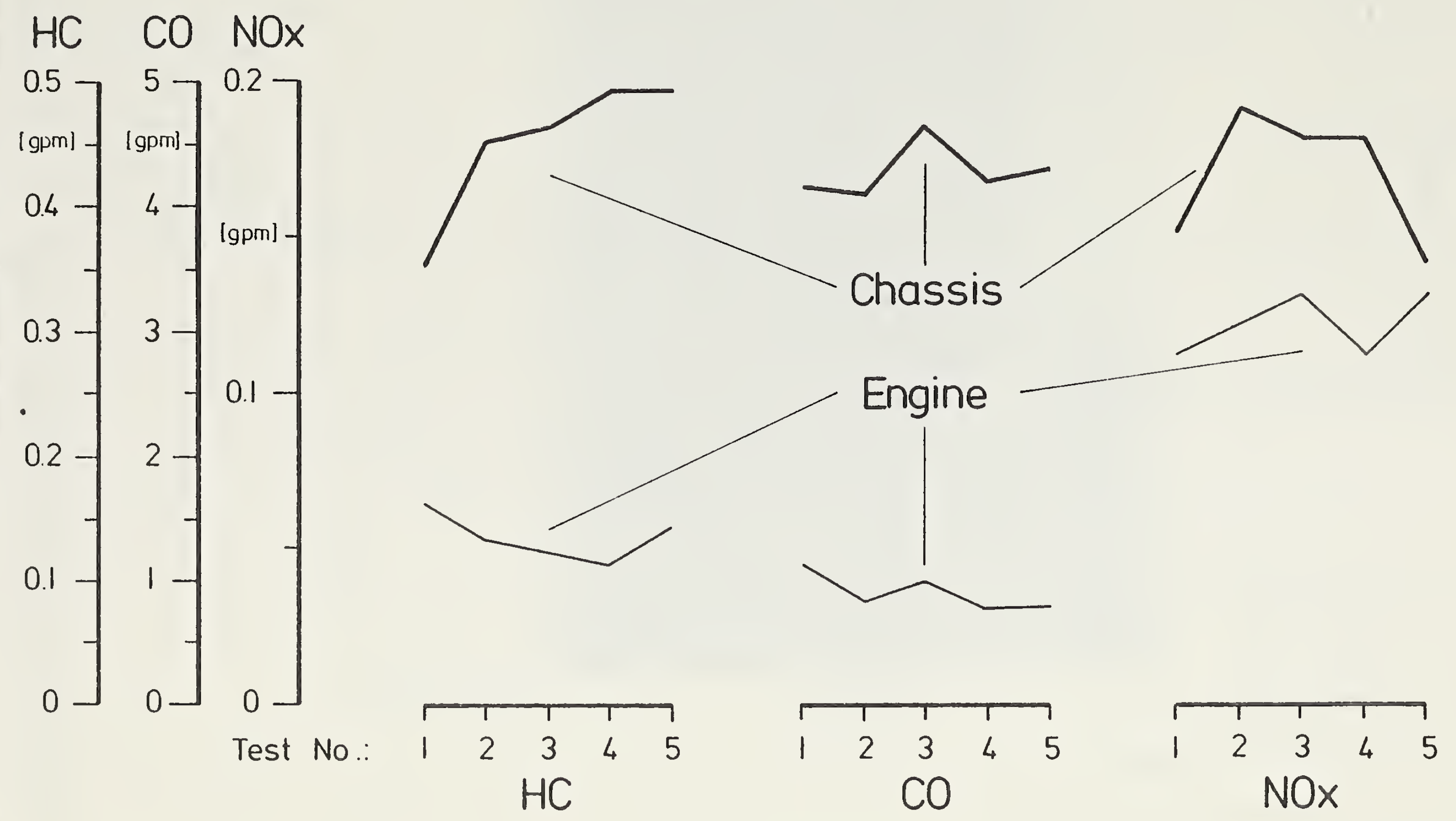


Fig.: 6. 2.1

Dilution Tunnel

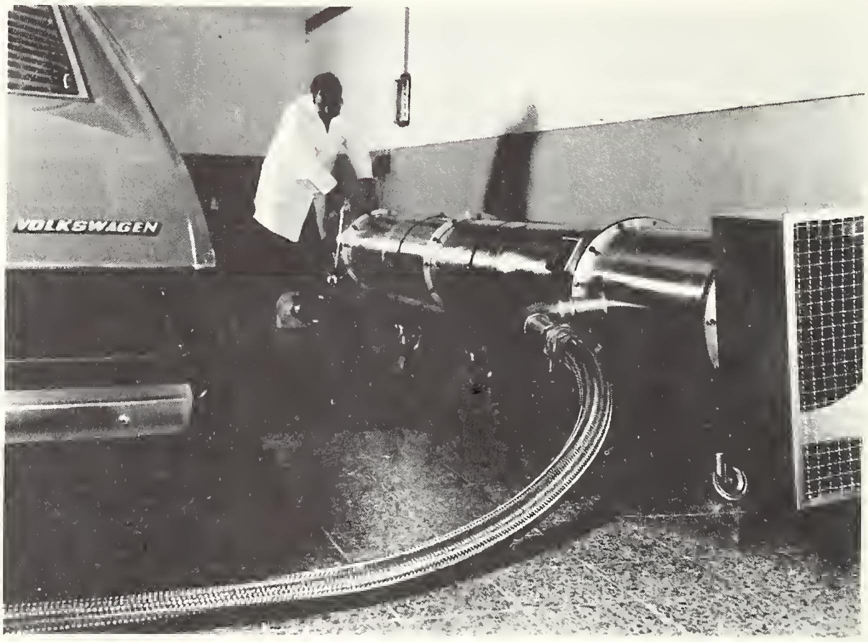


Fig.: 6.2.2

System of Taking Samples to Determine HCN and SO_4

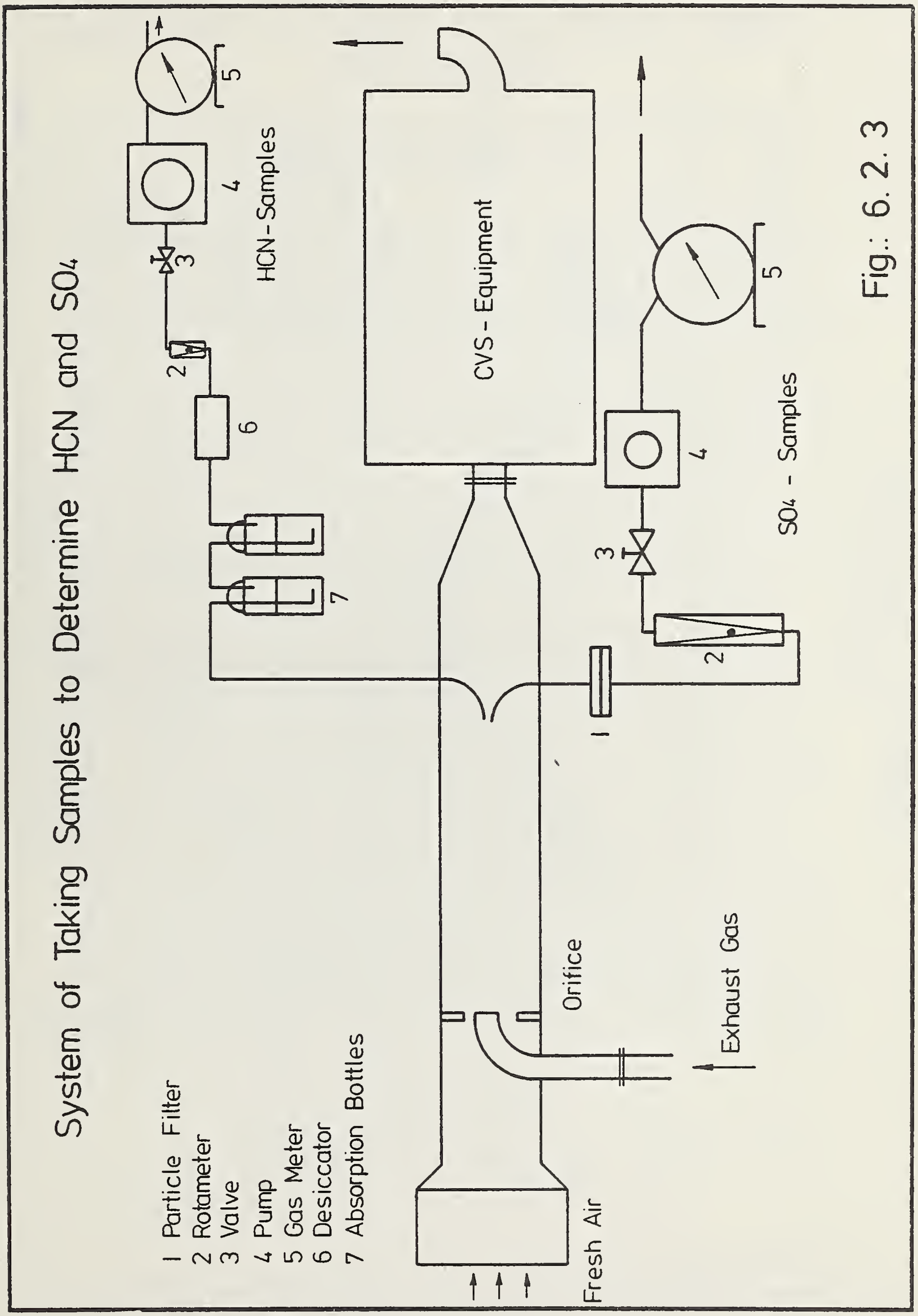
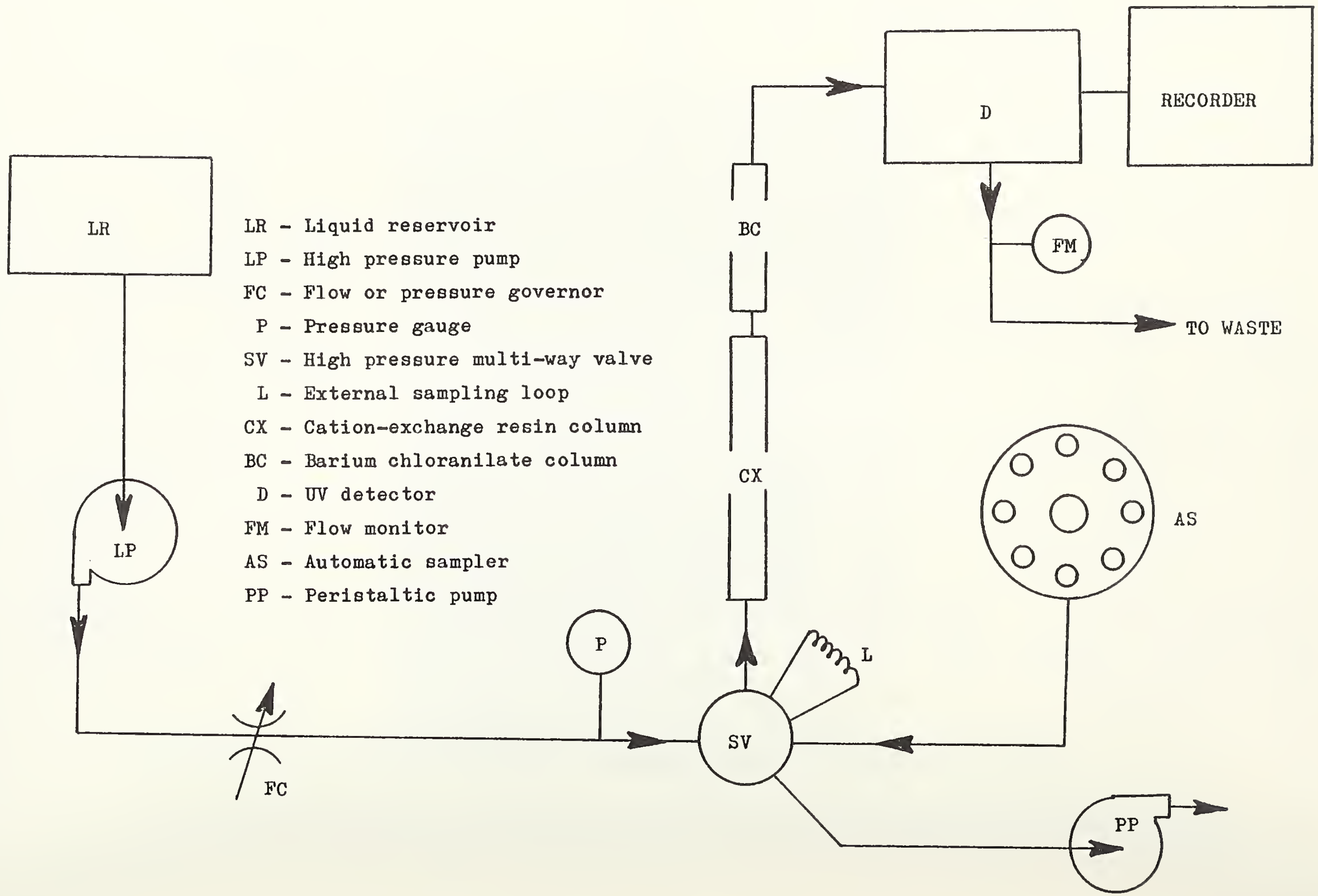


Fig.: 6. 2. 3



FLOW DIAGRAM OF AUTOMATED SULFATE INSTRUMENT

Fig.: 6.2.4

Noise Level vs. Progress During SAE J 986 A Drive-By Test

Modification Code 01

Transmission : Manual ⊗ Automatic ○

Speed : 4

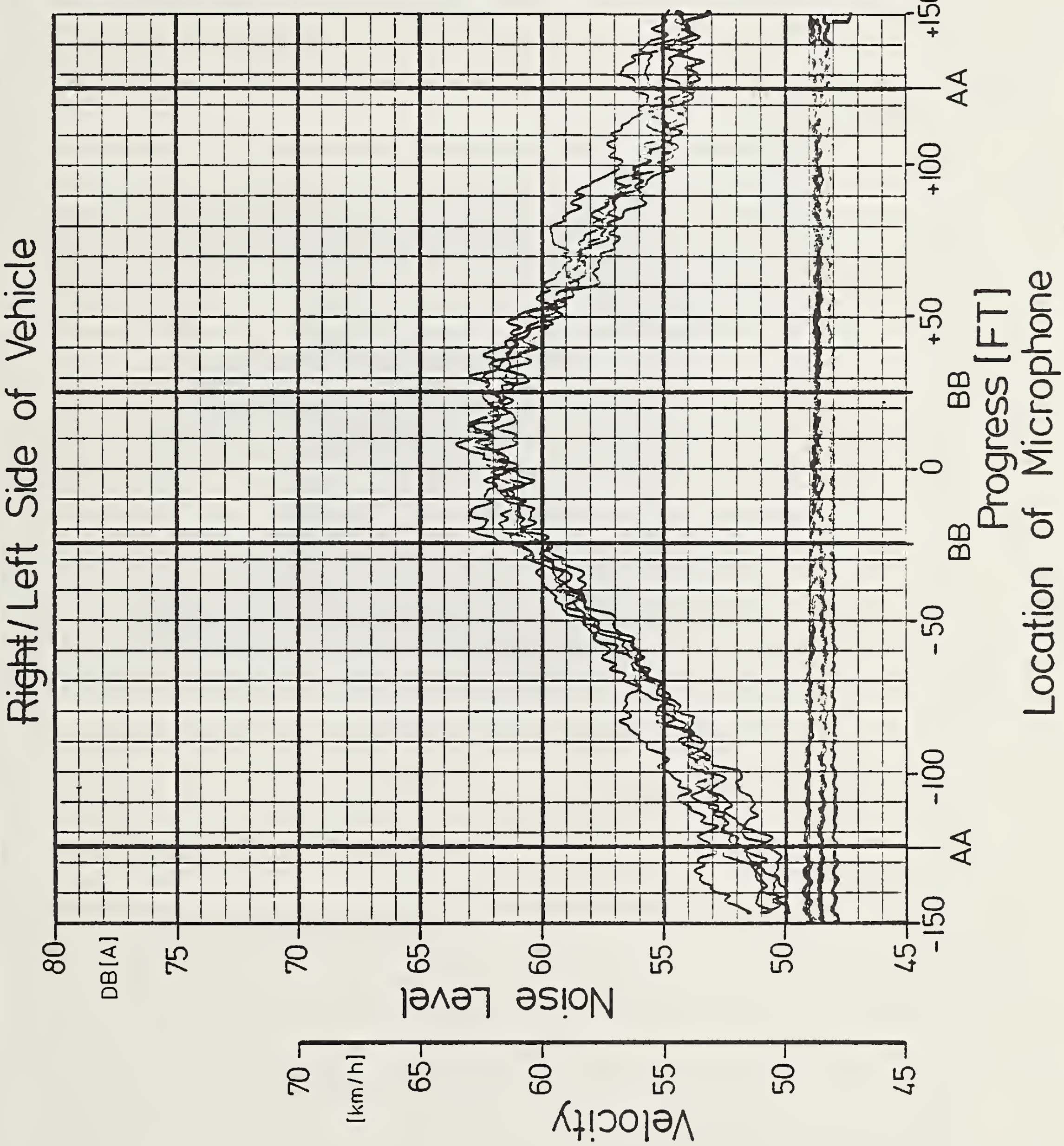


Fig.: 6.2.5

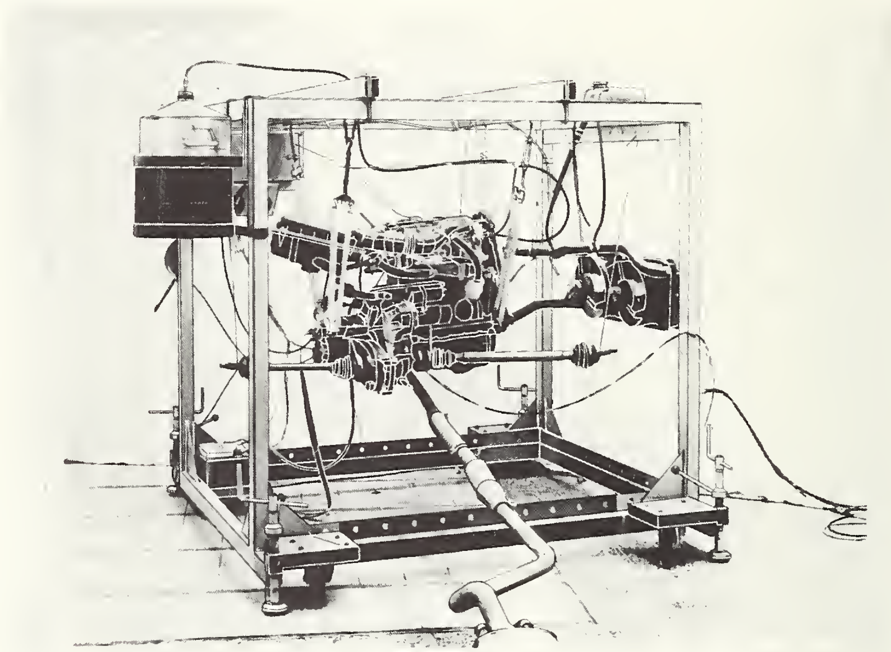


Fig.: 6.2.6

Vibration piston direction
~~crankshaft direction~~
~~transversal~~

Engine : 4 Zyl.; ~~1.6~~ 1.3 ltr displ.

Position No.: I

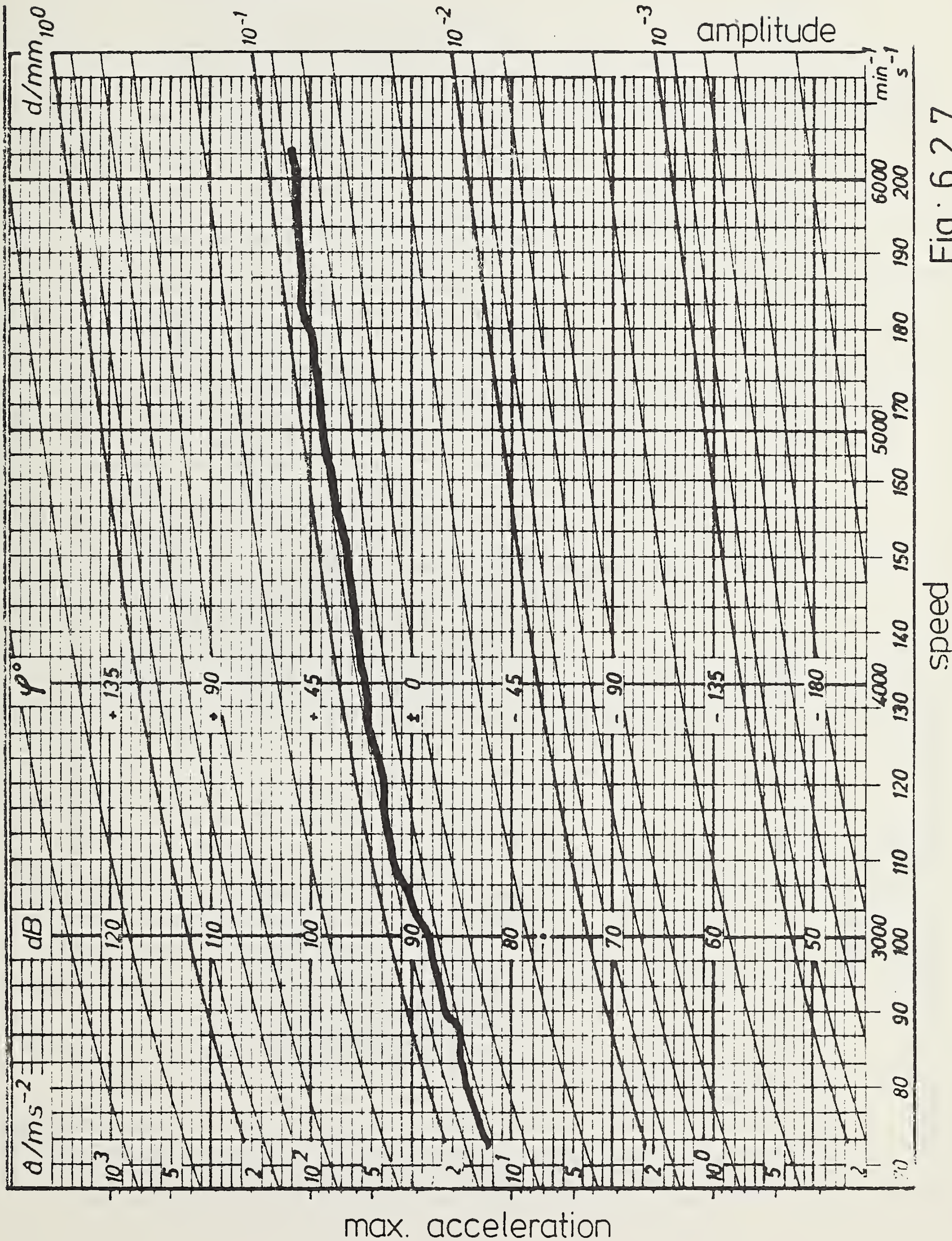
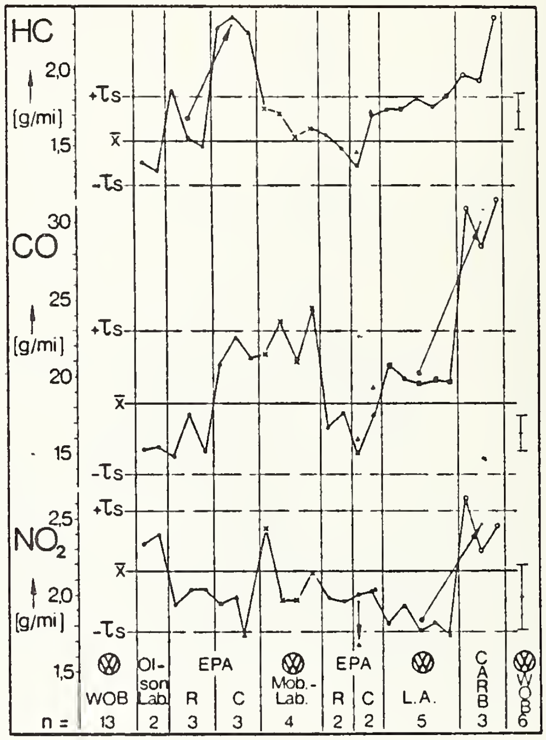


Fig.: 6.2.7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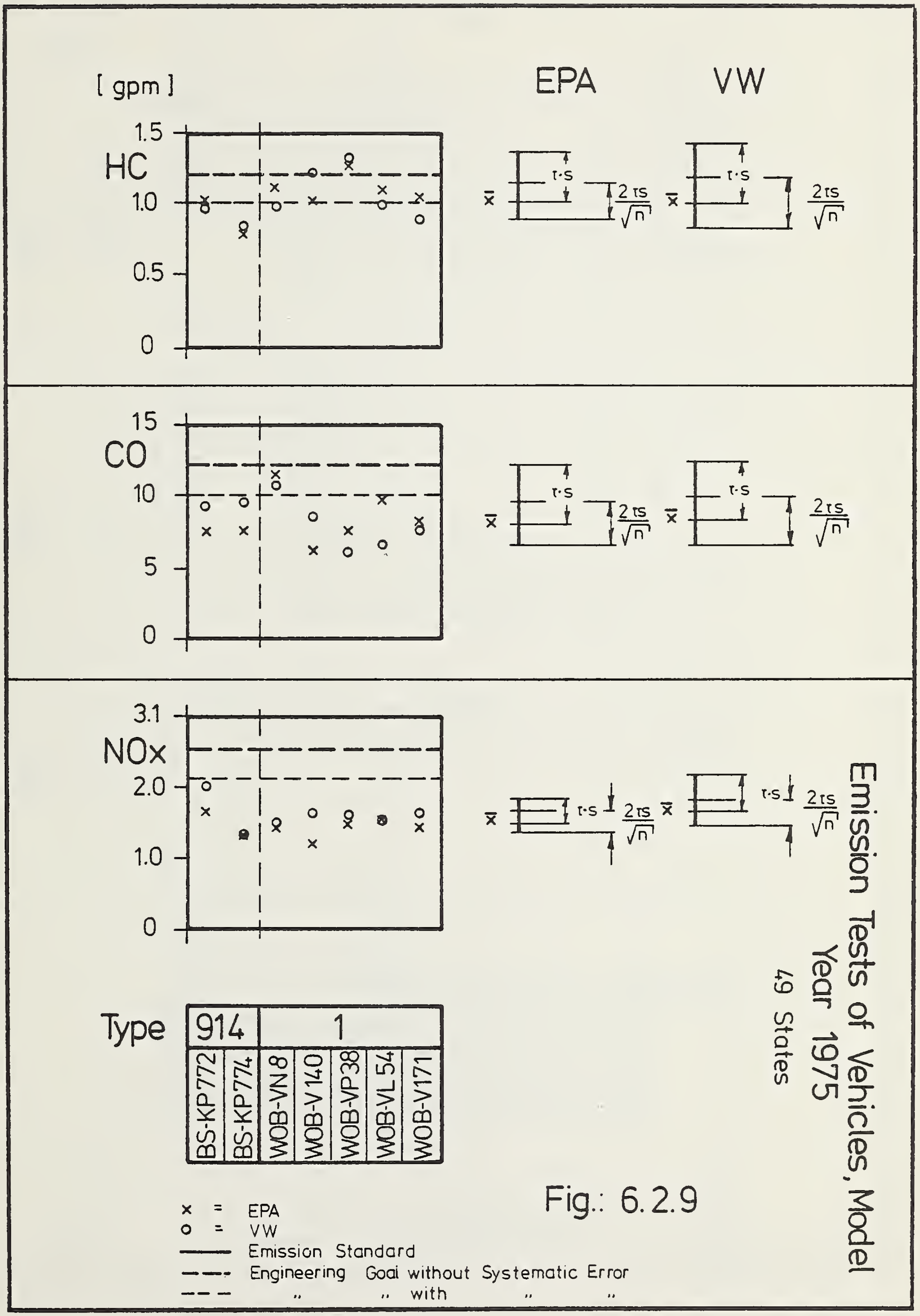
+τs
[g/mi]
1,5
x̄
-τs
30
CO
25
+τs
[g/mi]
20
x̄
15
-τs
+τs
2,5
NO_2
x̄
2,0
[g/mi]
-τs
1,5
Ol-son Lab.
EPA
R
C
Mob.-Lab.
EPA
R
C
WOB
L.A.
CARB
WOB
n =
13
2
3
3
4
2
2
5
3
6

Fig.: 6.2.8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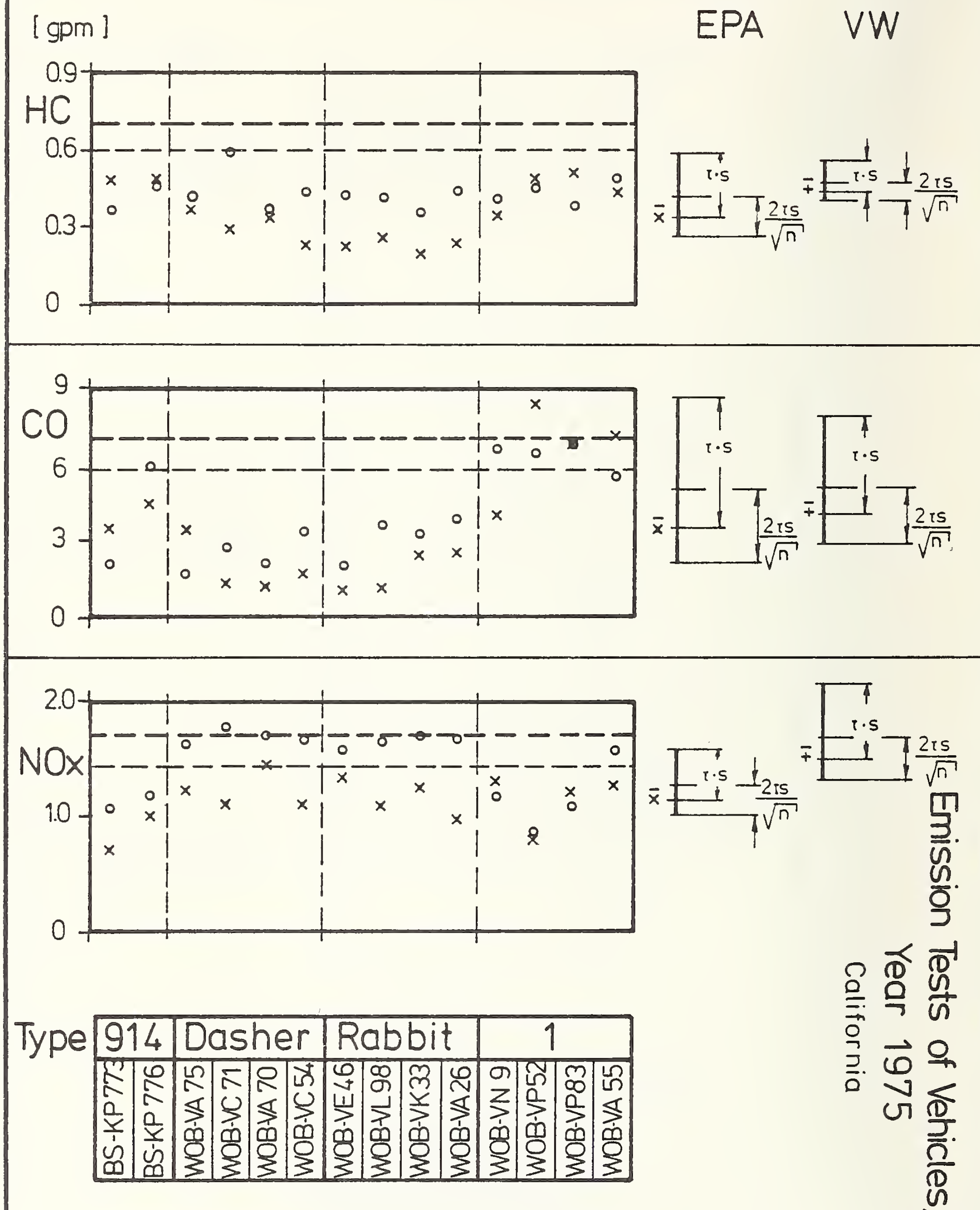
EPA
VW
HC
1.5
1.0
0.5
0
CO
15
10
5
0
NOx
3.1
2.0
1.0
0
$\bar{x}$
$t \cdot s$
$\frac{2ts}{\sqrt{n}}$
Type
914
1
BS-KP772
BS-KP774
WOB-VN 8
WOB-V140
WOB-VP38
WOB-VL 54
WOB-V171
x = EPA
o = VW
Emission Standard
Engineering Goal without Systematic Error
" " with " "
Emission Tests of Vehicles, Model Year 1975
49 States

Fig.: 6.2.9



× = EPA

o = VW

—— Emission Standard

— — Engineering Goal without Systematic Error

— — " " with " "

Emission Tests of Vehicles, Model Year 1975
California

Fig.: 6. 2. 10

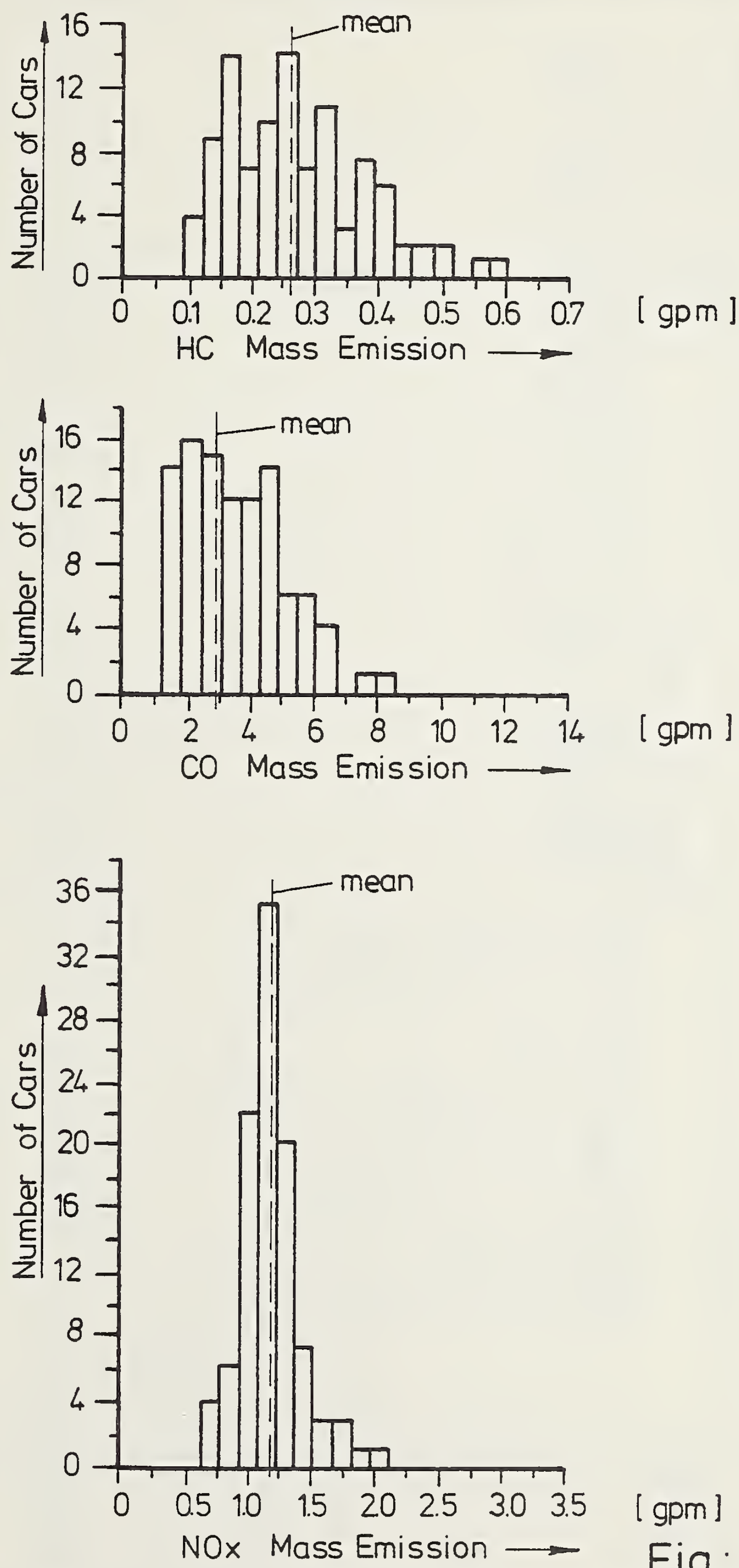

mean
Number of Cars
16
12
8
4
0
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7
[gpm]
HC Mass Emission
mean
Number of Cars
16
12
8
4
0
0 2 4 6 8 10 12 14
[gpm]
CO Mass Emission
mean
Number of Cars
36
32
28
24
20
16
12
8
4
0
0 0.5 1.0 1.5 2.0 2.5 3.0 3.5
[gpm]
NOx Mass Emission

Fig.: 6.2.11

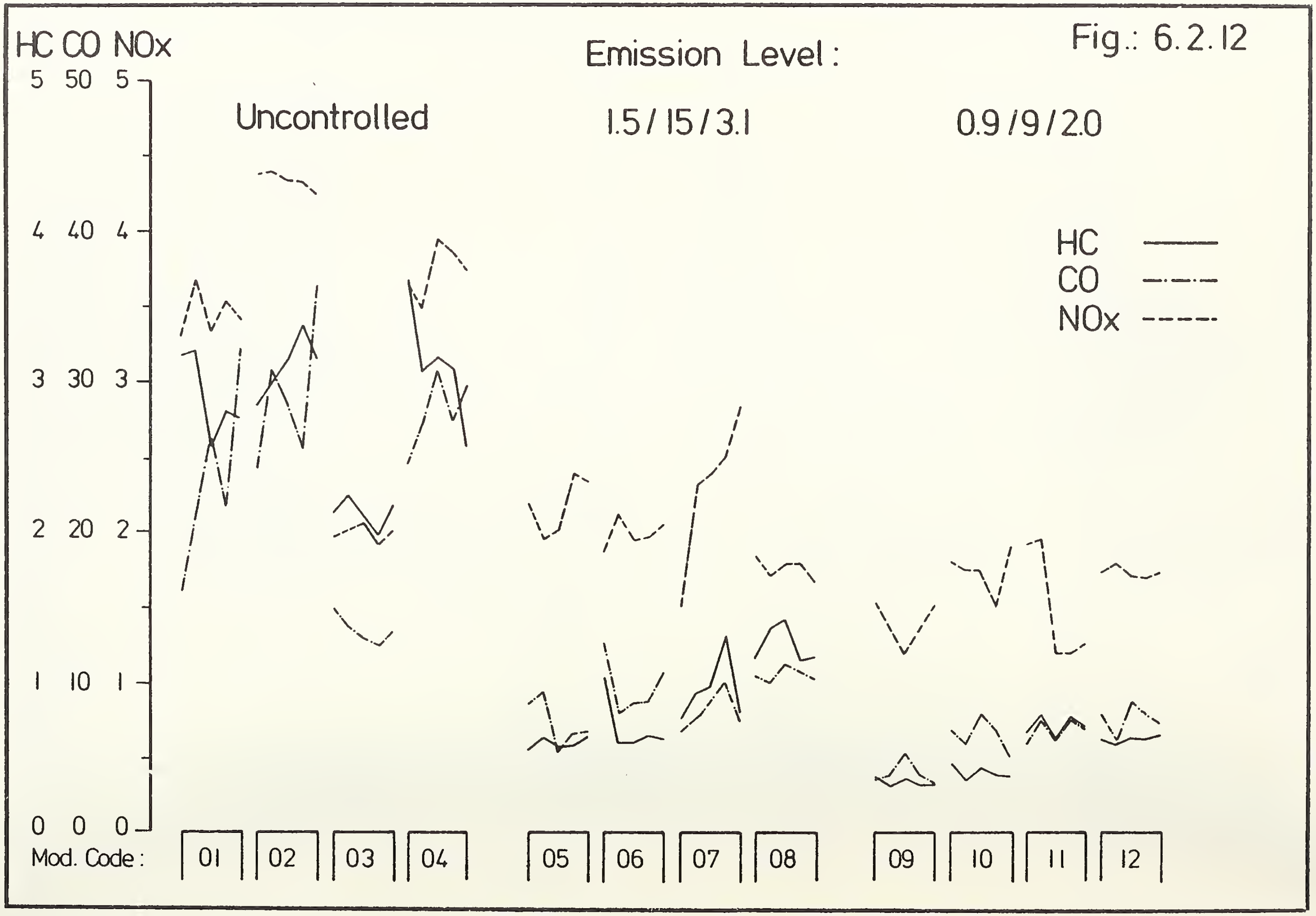


Fig.: 6.2.12
Emission Level:
Uncontrolled
1.5/15/3.1
0.9/9/2.0
HC CO NOx
5 50 5
4 40 4
3 30 3
2 20 2
1 10 1
0 0 0
Mod. Code:
01
02
03
04
05
06
07
08
09
10
11
12
HC
CO
NOx

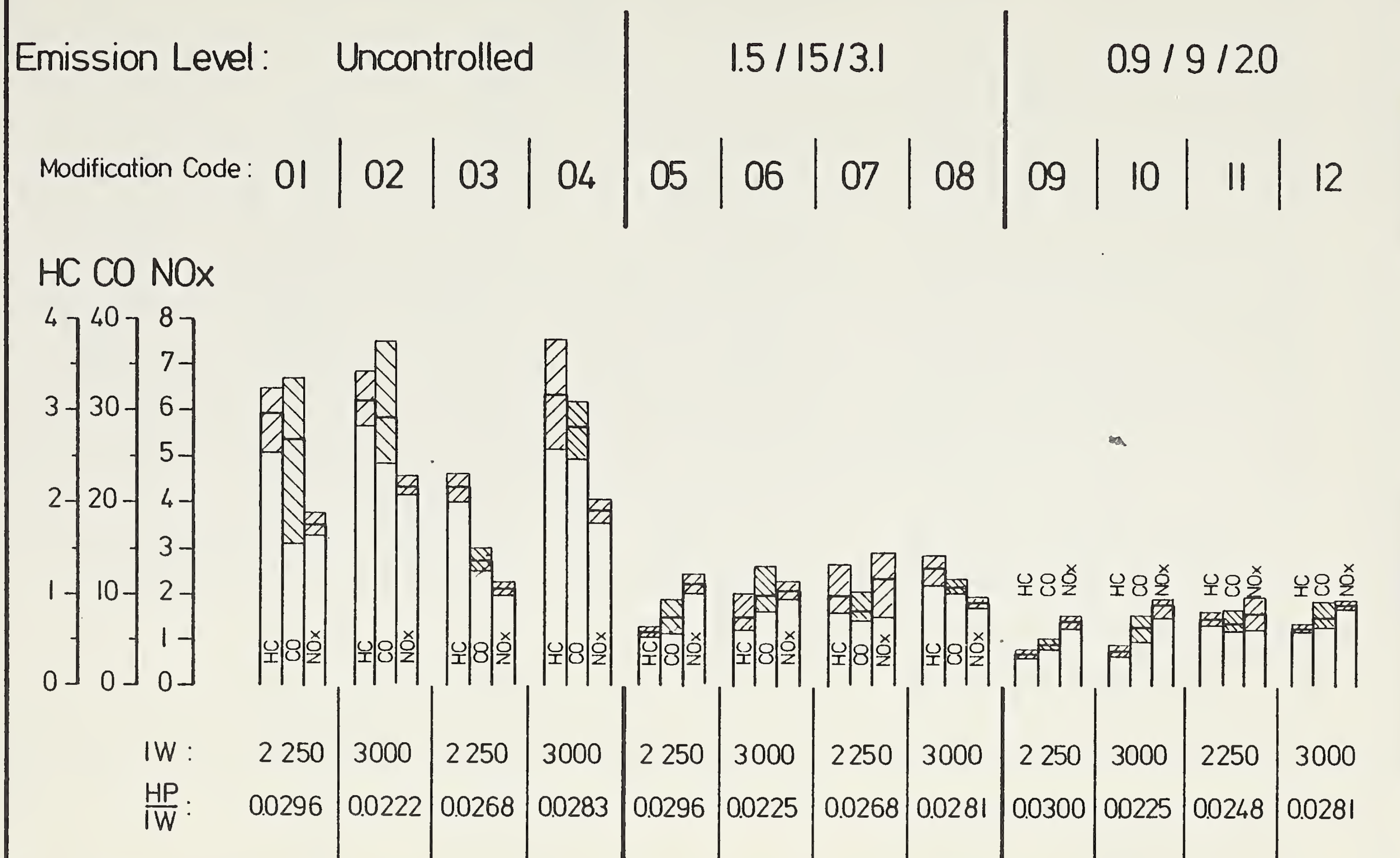
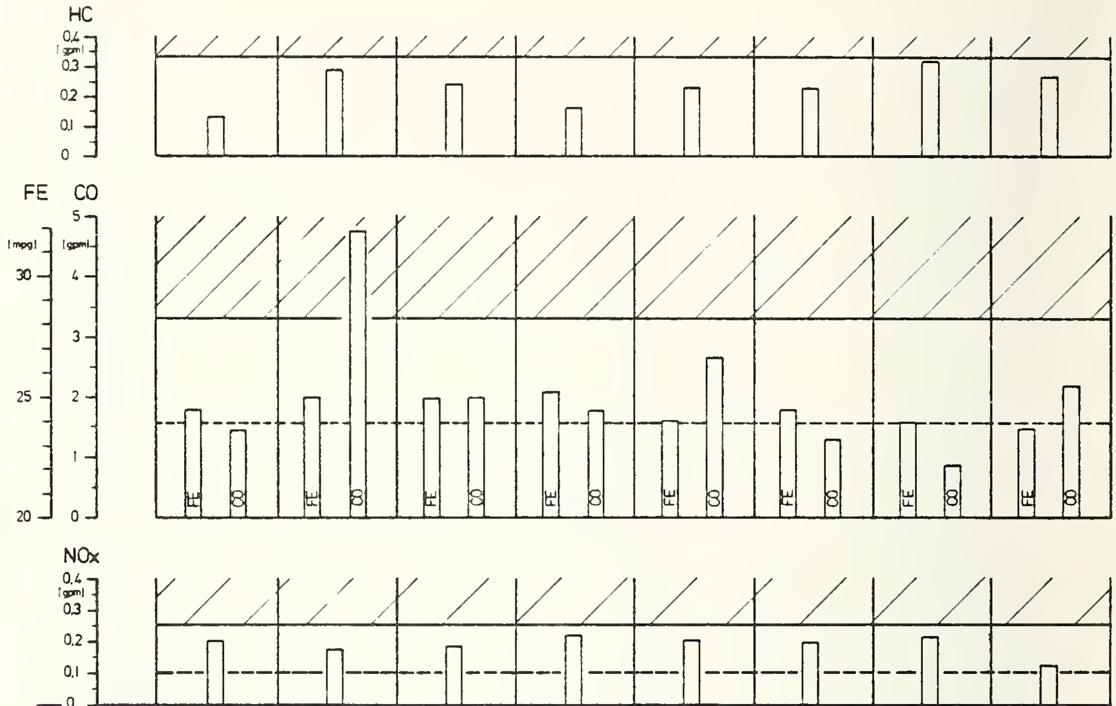

Emission Level :
Uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
Modification Code :
01
02
03
04
05
06
07
08
09
10
11
12
HC CO NOx
4
3
2
1
0
40
30
20
10
0
8
7
6
5
4
3
2
1
0
HC
CO
NOx
IW :
2 250
3000
2 250
3000
2 250
3000
2250
3000
2 250
3000
2250
3000
HP/IW :
0.0296
0.0222
0.0268
0.0283
0.0296
0.0225
0.0268
0.0281
0.0300
0.0225
0.0248
0.0281

Fig.: 6. 2. 13

Adjustment Variations for Modification Code 17



Test-No	18	19	20	21	23	24	25	26
	withoud cold-start injection at + 20°C	→	→	→	→	→	→	→
	warm-up regulator with altitude-sensor	→	→	→	→	warm-up regulator without altitude sensor (as serial)	→	→
	Starting Control pressure (20°C) 2.0 bar (serial)	→	→	→	→	→	→ 2.5 bar (leaner)	→ 2.0 bar
	Control pressure hot engine 3.55 bar (serial)	→	→	→	→	→	→	→ 3.0 bar (richer)
	150 ltr/min S.A. after 160sec switched from TWC to OXCAT	150 ltr/min S.A. from the beginning to OXCAT	150 ltr/min S.A. after 40 sec switched from TWC to OXCAT	→ after 50 sec →	→ after 60 sec →	→	→	→
	Idle spark timing 3° ATDC	→	→	→	→	→	→	→
			40 sec after start without spark advance diaphragm	50 sec after →	90 sec after →	→	→	→

Fig.: 6. 2. 14

Adjustment Variations for Mod. Code 17

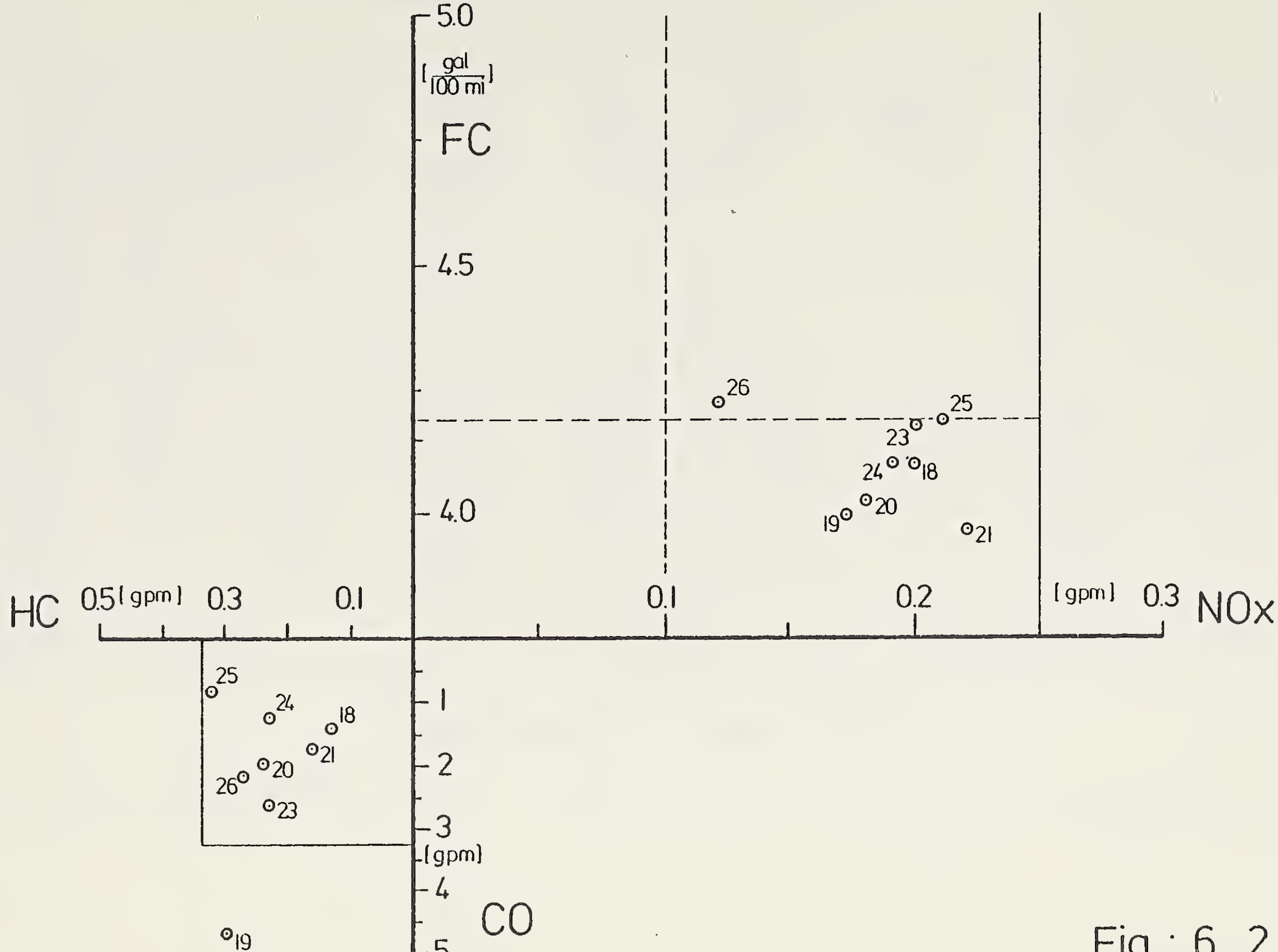
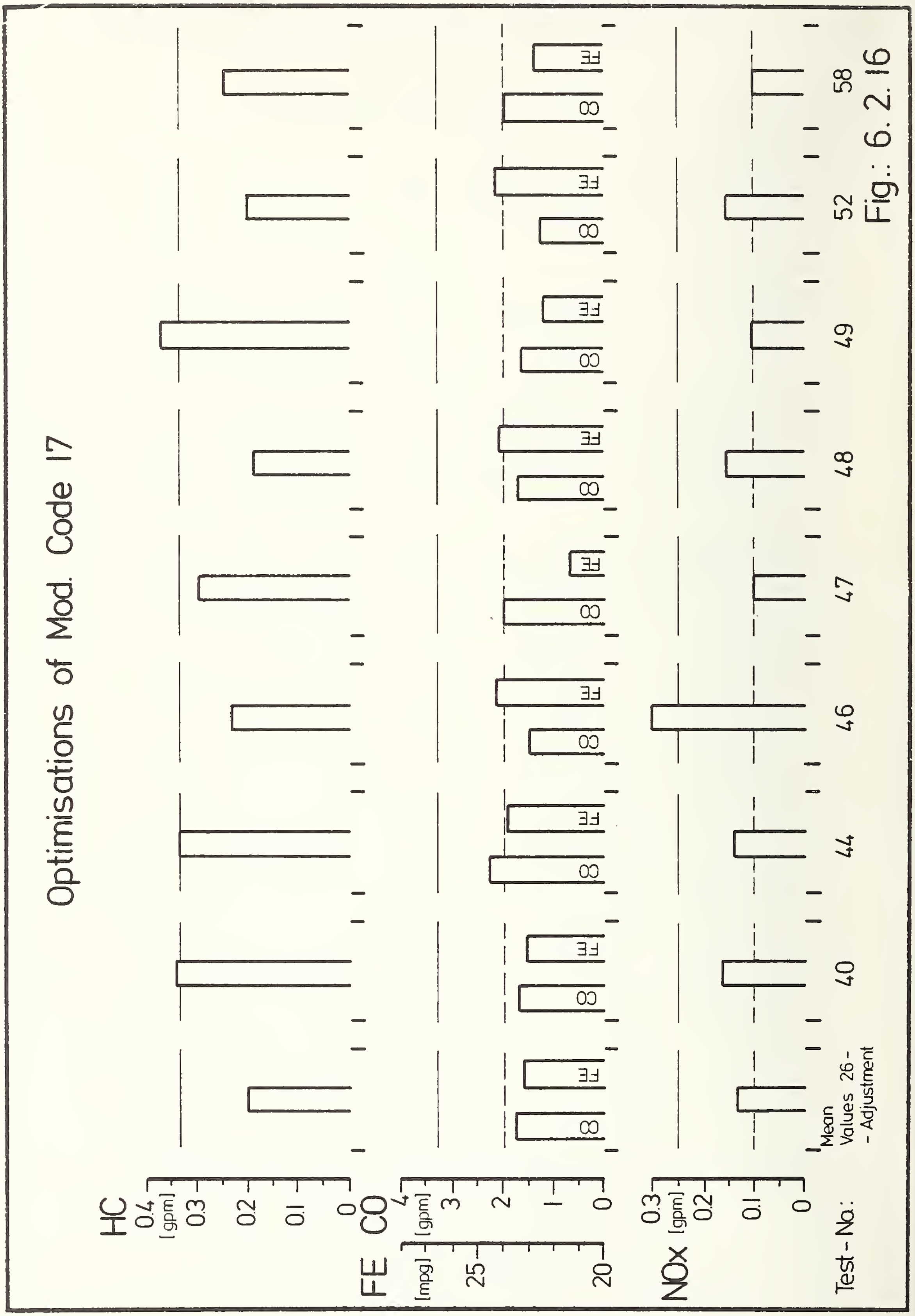


Fig.: 6. 2. 15



Optimisations of Mod. Code 17
HC [gpm] 0.4 0.3 0.2 0.1 0
FE [mpg] 25 20
CO [gpm] 4 3 2 1 0
NOx [gpm] 0.3 0.2 0.1 0
CO FE
Test-No.:
Mean Values 26-Adjustment
40 44 46 47 48 49 52 58

Fig.: 6. 2. 16

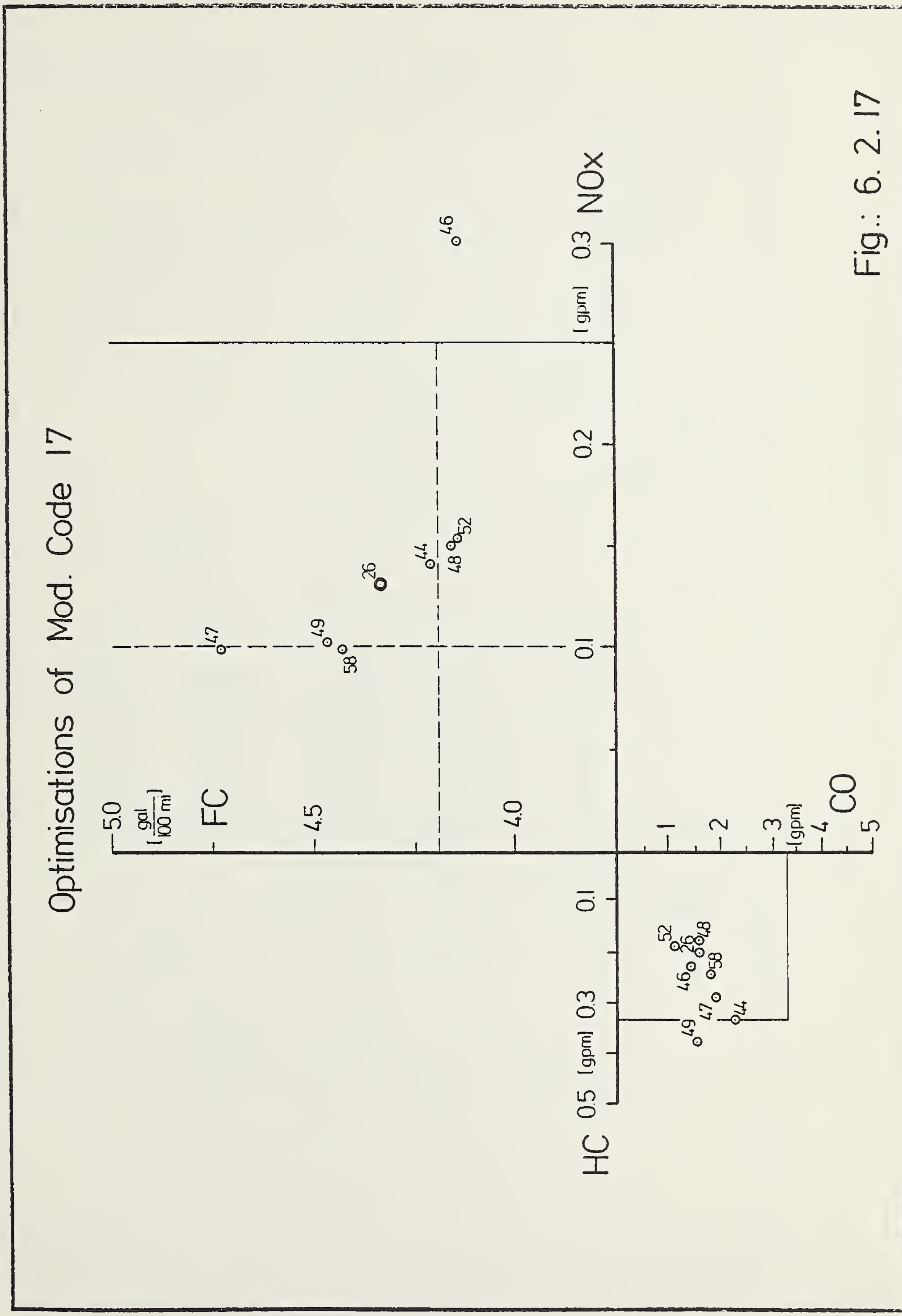

Optimisations of Mod. Code 17
FC
[gal/100 mi]
5.0
4.5
4.0
NOx
[gpm]
0.1
0.2
0.3
CO
[gpm]
1
2
3
4
5
HC
[gpm]
0.1
0.3
0.5
47
49
58
26
44
48
52
46

Fig.: 6. 2.17

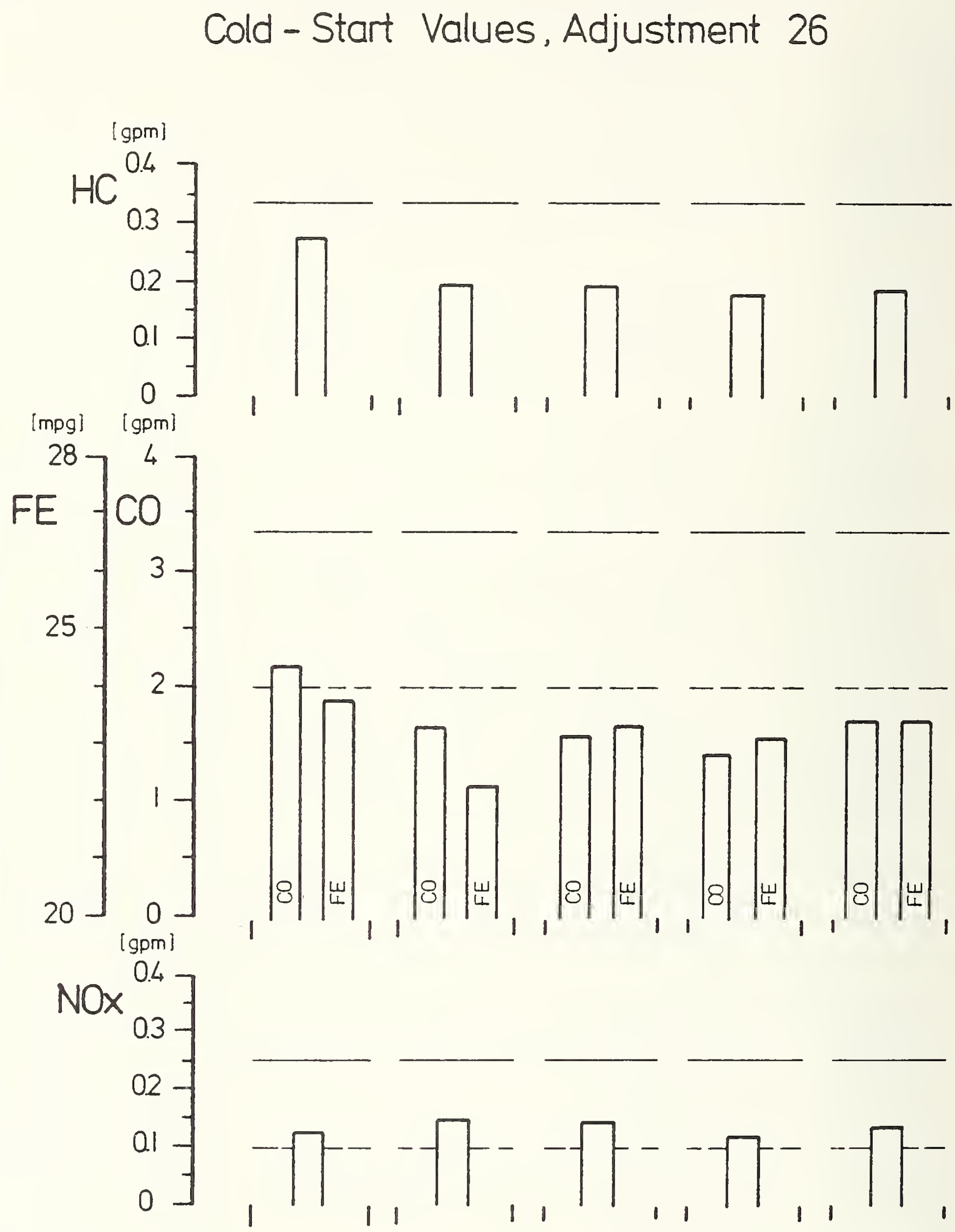

Cold - Start Values, Adjustment 26
[gpm]
HC
0.4
0.3
0.2
0.1
0
[mpg]
[gpm]
FE
CO
28
25
20
4
3
2
1
0
CO
FE
[gpm]
NOx
0.4
0.3
0.2
0.1
0

Fig.: 6. 2. 18

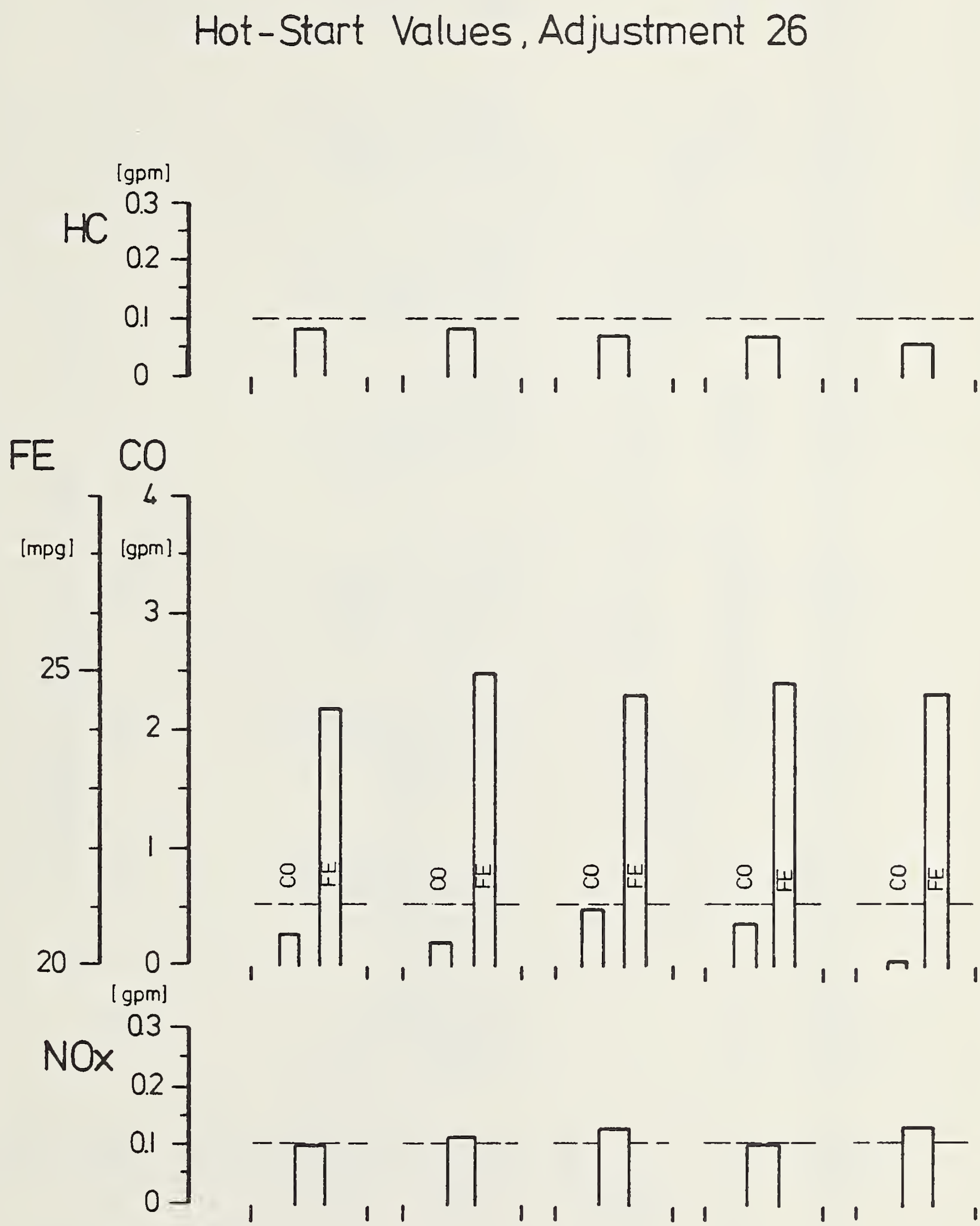

Hot-Start Values, Adjustment 26
HC
[gpm]
0.3
0.2
0.1
0
FE
CO
[mpg]
[gpm]
4
3
25
2
1
20
0
CO
FE
NOx
[gpm]
0.3
0.2
0.1
0

Fig.: 6.2.19

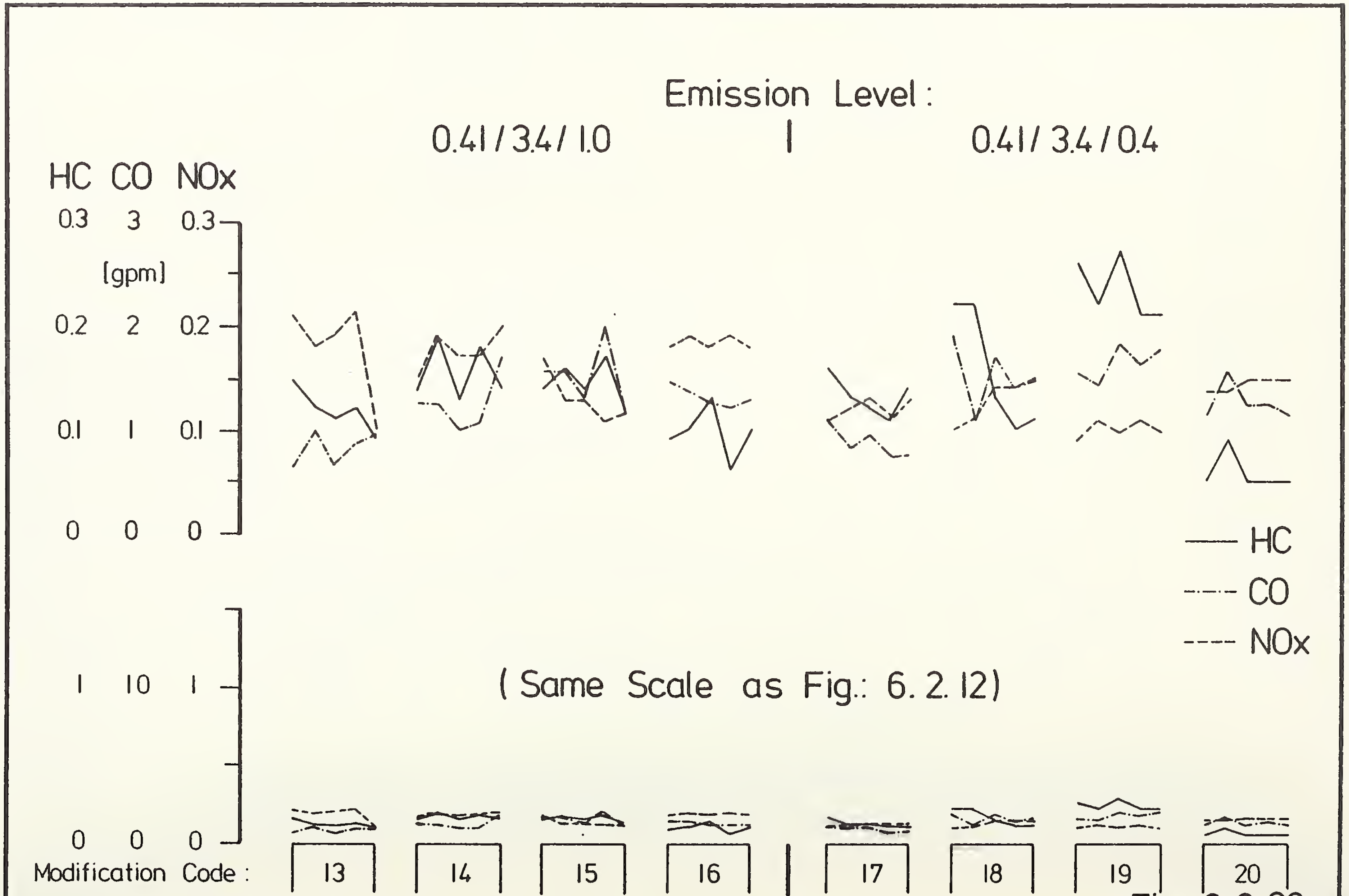

Emission Level :
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4
HC CO NOx
0.3 3 0.3
[gpm]
0.2 2 0.2
0.1 1 0.1
0 0 0
1 10 1
0 0 0
(Same Scale as Fig.: 6. 2. 12)
HC
CO
NOx
Modification Code :
13
14
15
16
17
18
19
20

Fig.: 6. 2. 20

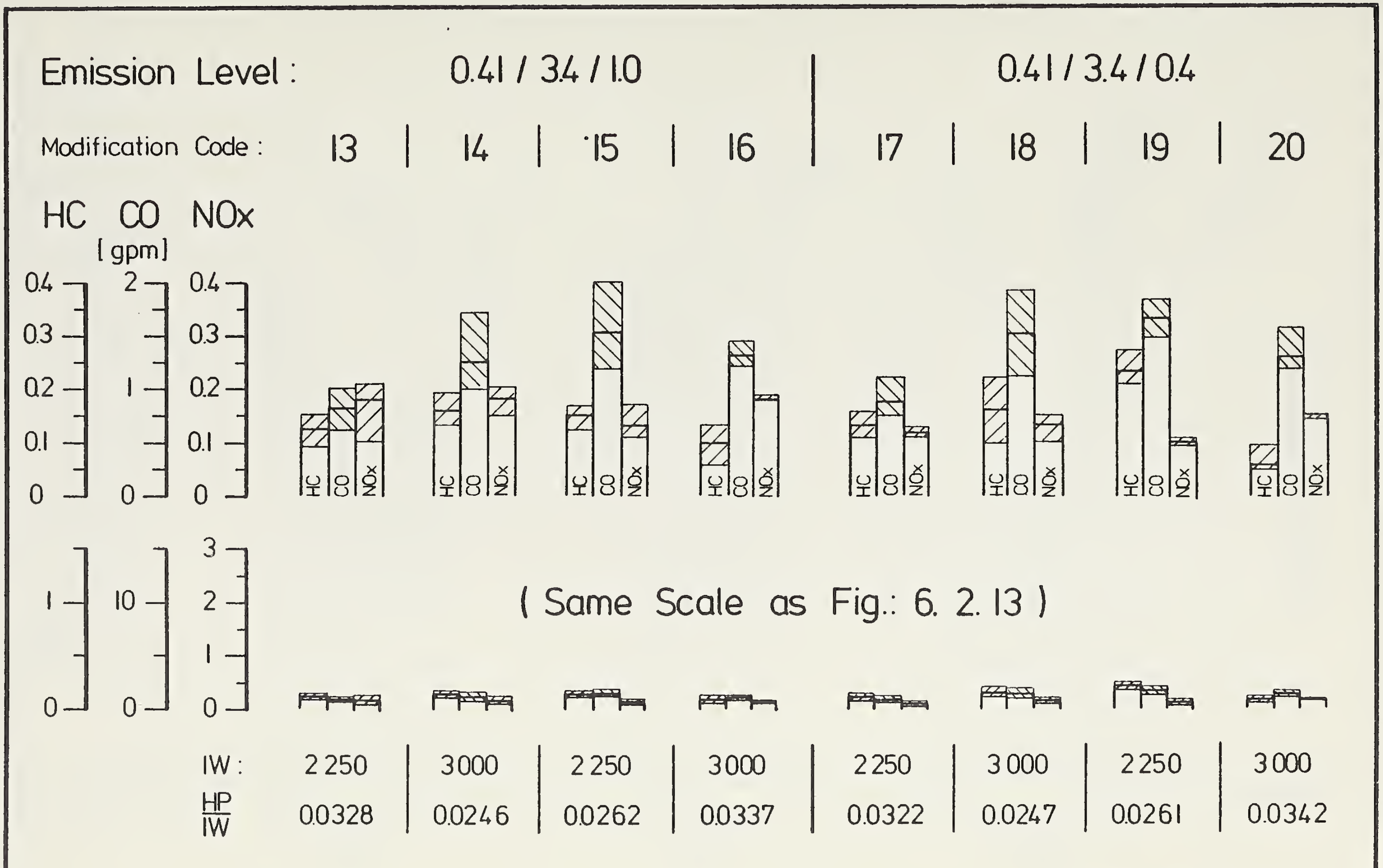
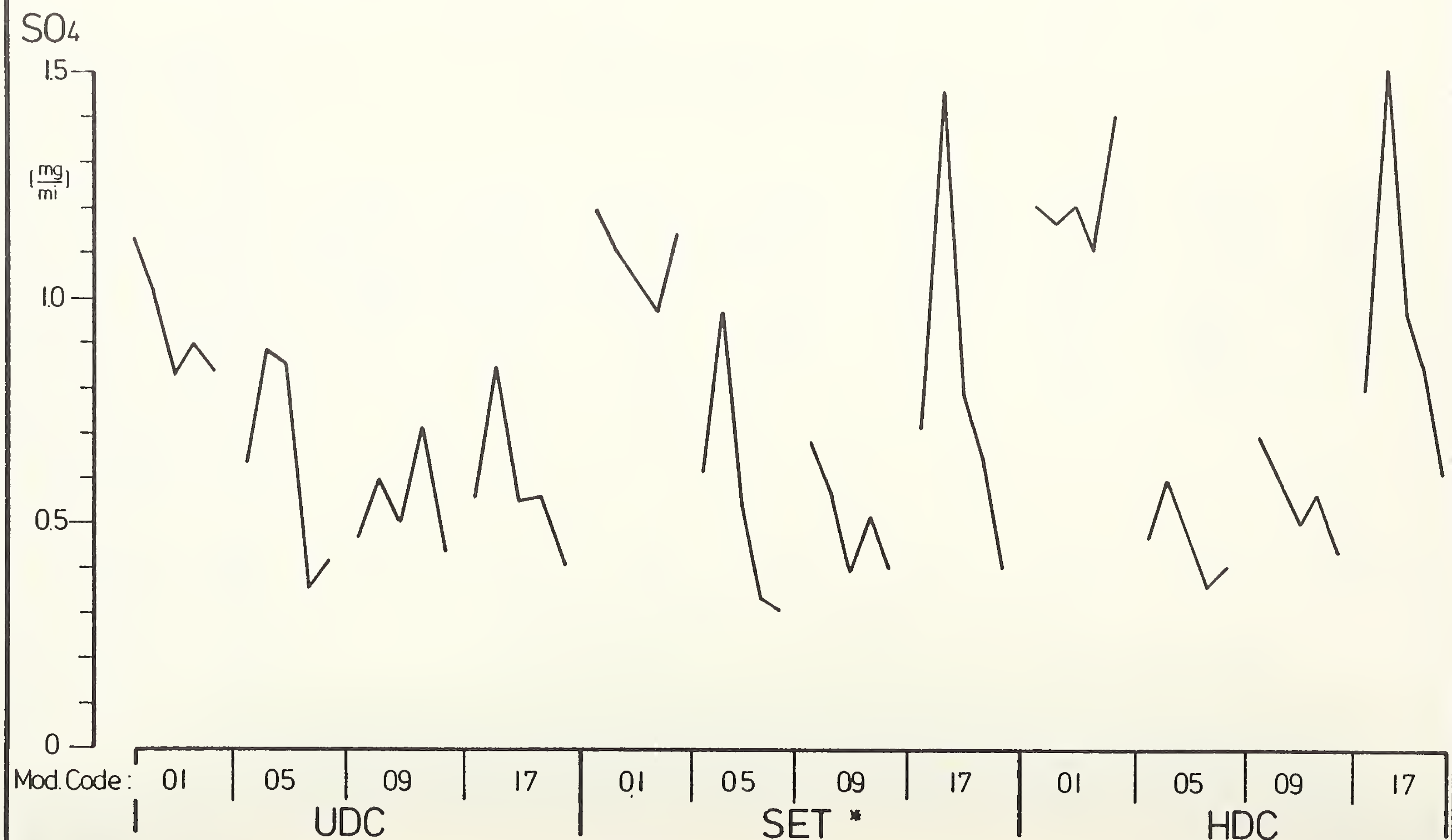

Emission Level :
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4
Modification Code :
13
14
15
16
17
18
19
20
HC CO NOx
[gpm]
0.4 0.3 0.2 0.1 0
2 1 0
0.4 0.3 0.2 0.1 0
HC CO NOx
1 0
10 0
3 2 1 0
(Same Scale as Fig.: 6. 2. 13)
IW :
2250 3000 2250 3000 2250 3000 2250 3000
HP/IW
0.0328 0.0246 0.0262 0.0337 0.0322 0.0247 0.0261 0.0342

Fig.: 6. 2. 21



* Sulphate Emission Test

Fig.: 6. 2. 22

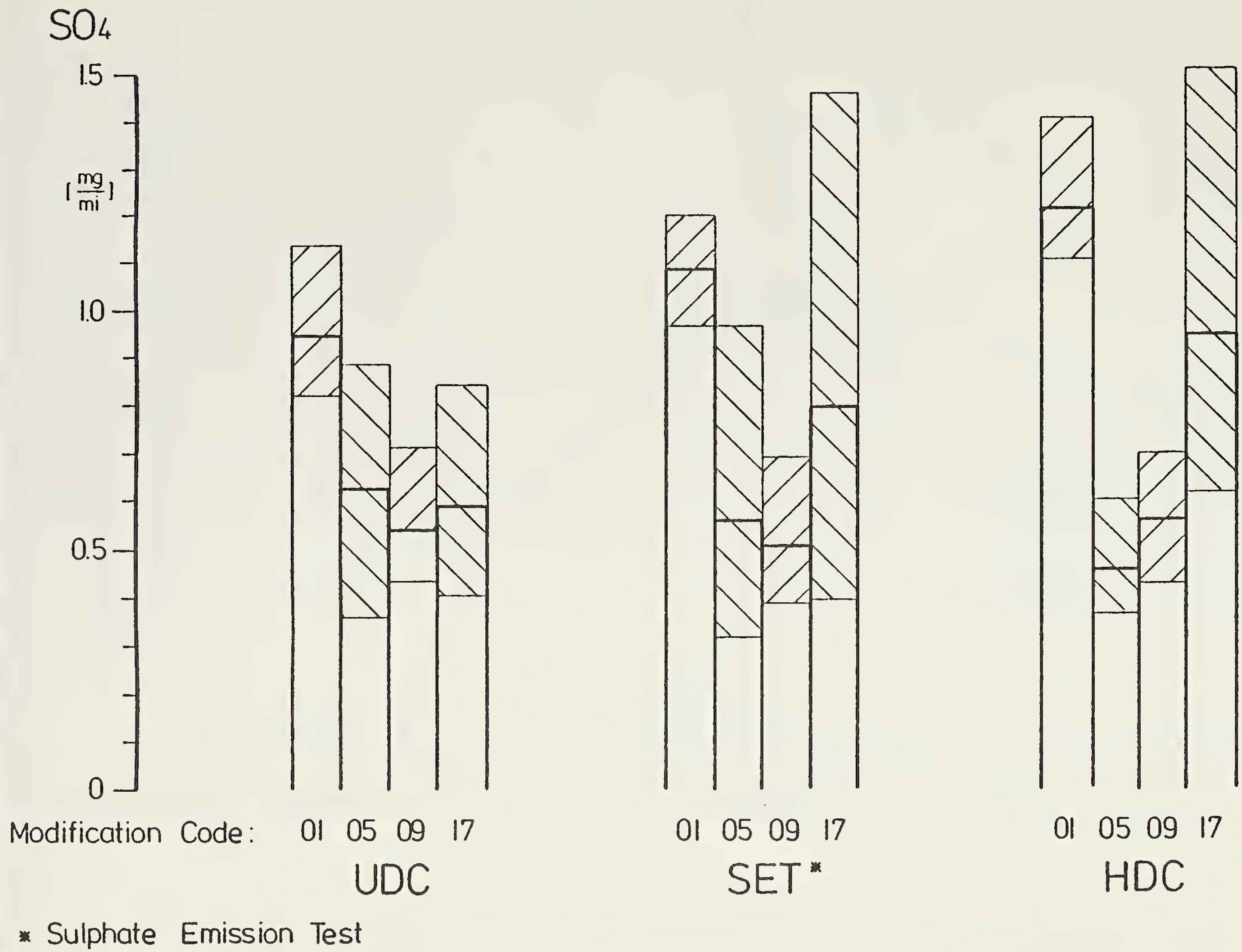

SO4
1.5
1.0
0.5
0
[mg/mi]
Modification Code: 01 05 09 17
01 05 09 17
01 05 09 17
UDC
SET *
HDC
* Sulphate Emission Test

Fig.: 6. 2. 23

Conversion of SO_2 to SO_3 by the DEGUSSA Three-Way-Catalyst
OM 721

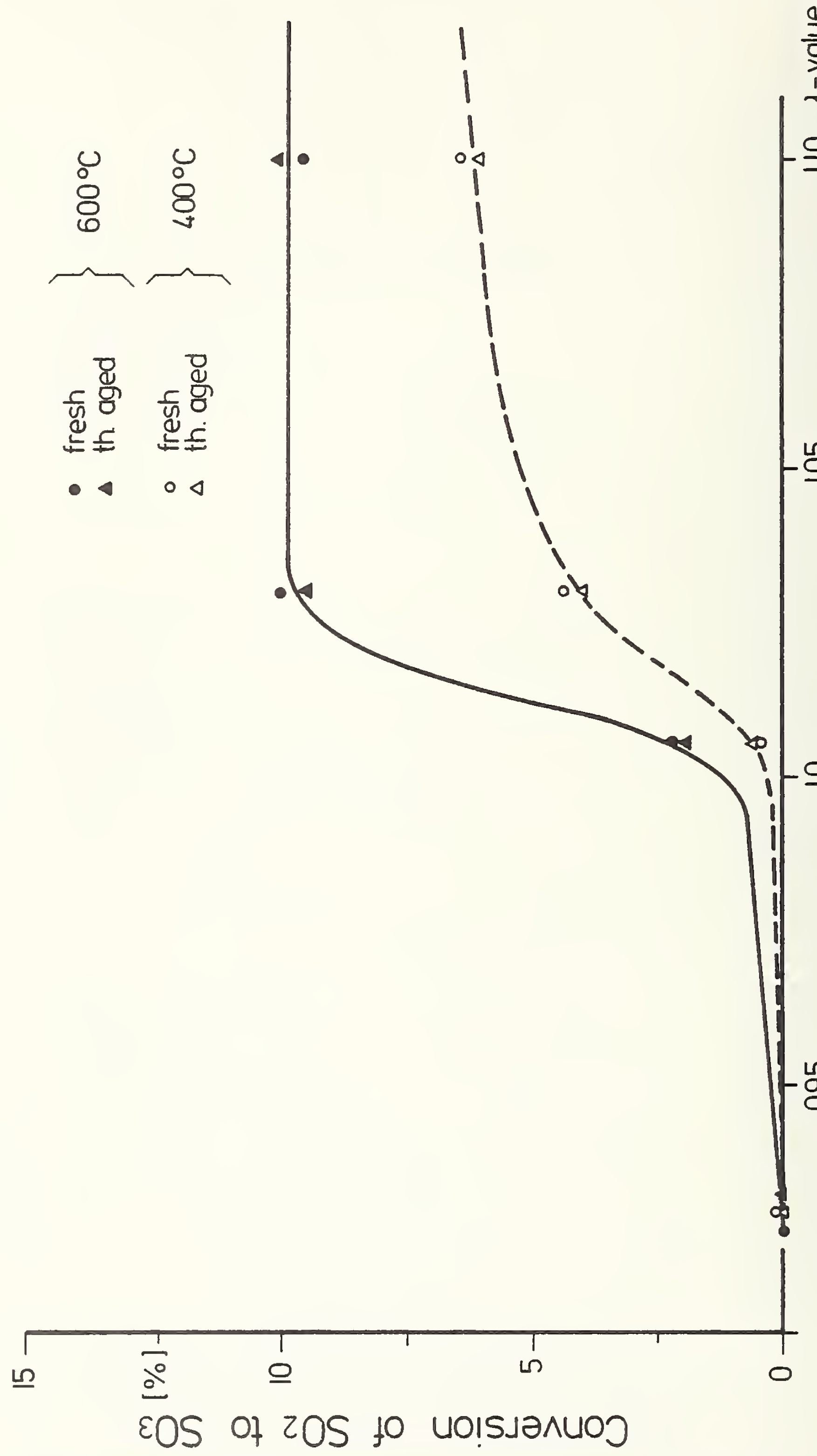


Fig.: 6. 2. 24

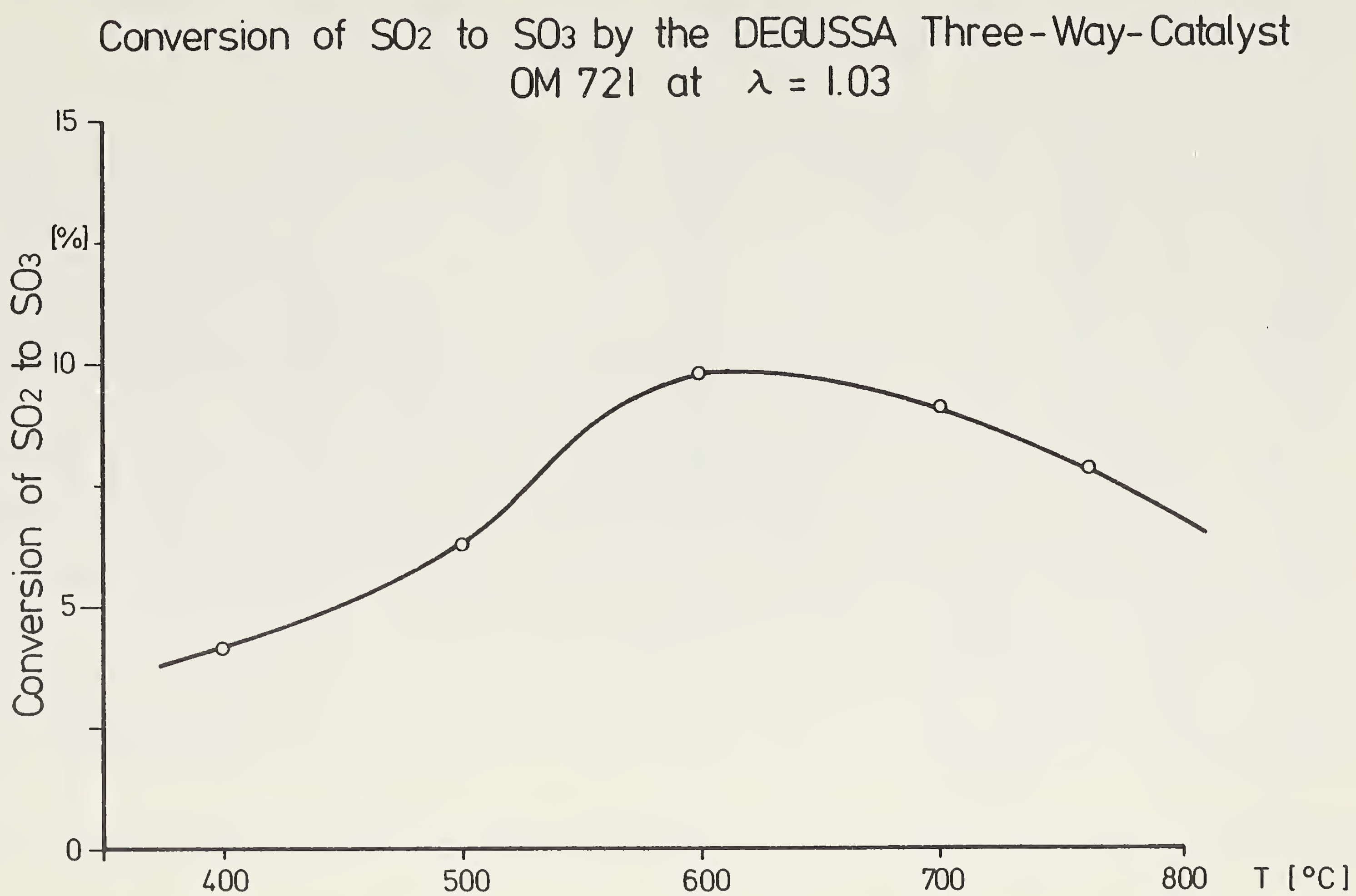
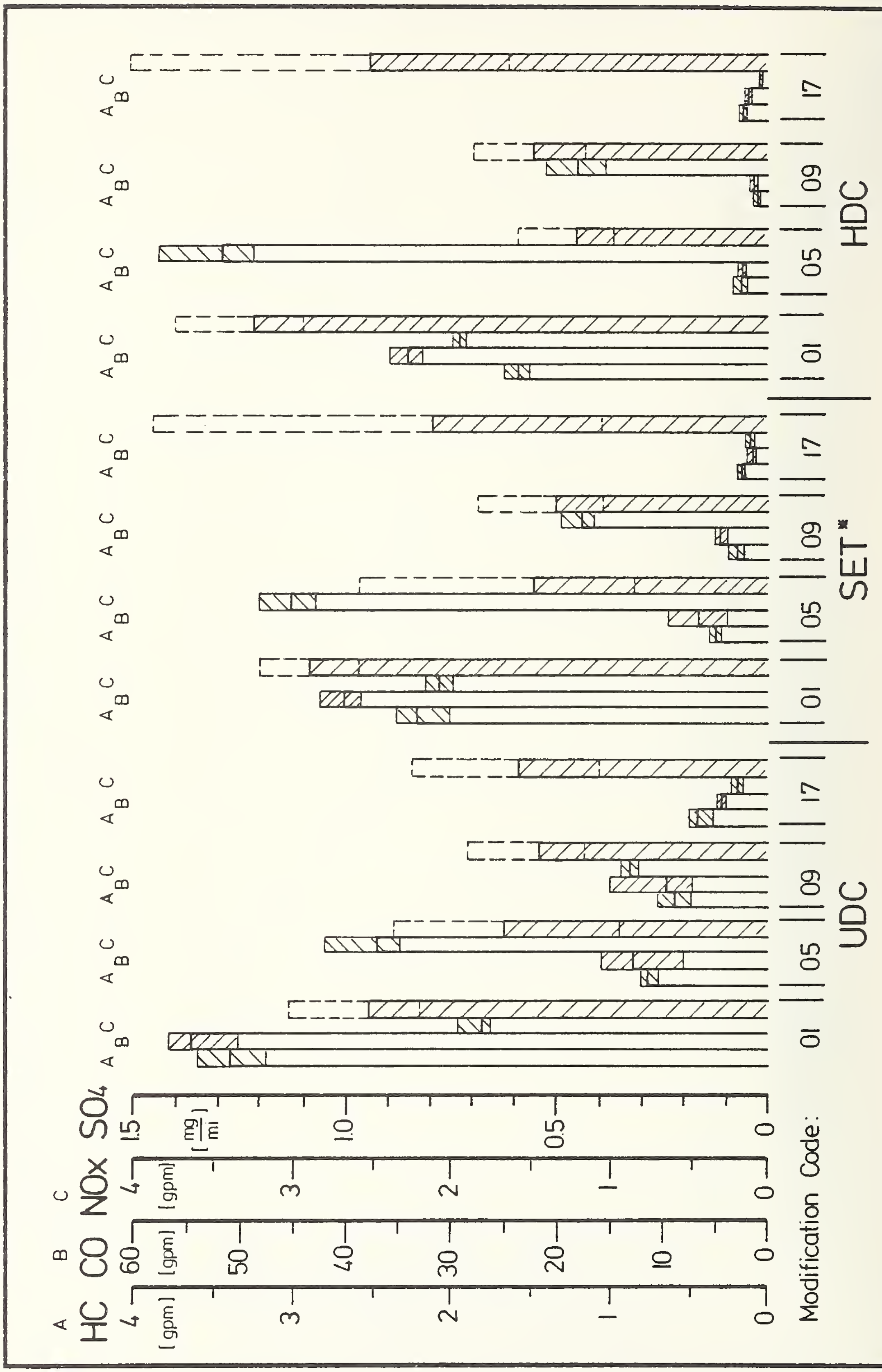

Conversion of SO_2 to SO_3 by the DEGUSSA Three-Way-Catalyst
OM 721 at λ = 1.03
Conversion of SO_2 to SO_3
[%]
15
10
5
0
400
500
600
700
800
T [°C]

Fig.: 6. 2. 25



* Sulphate Emission Test

Fig.: 6. 2. 26

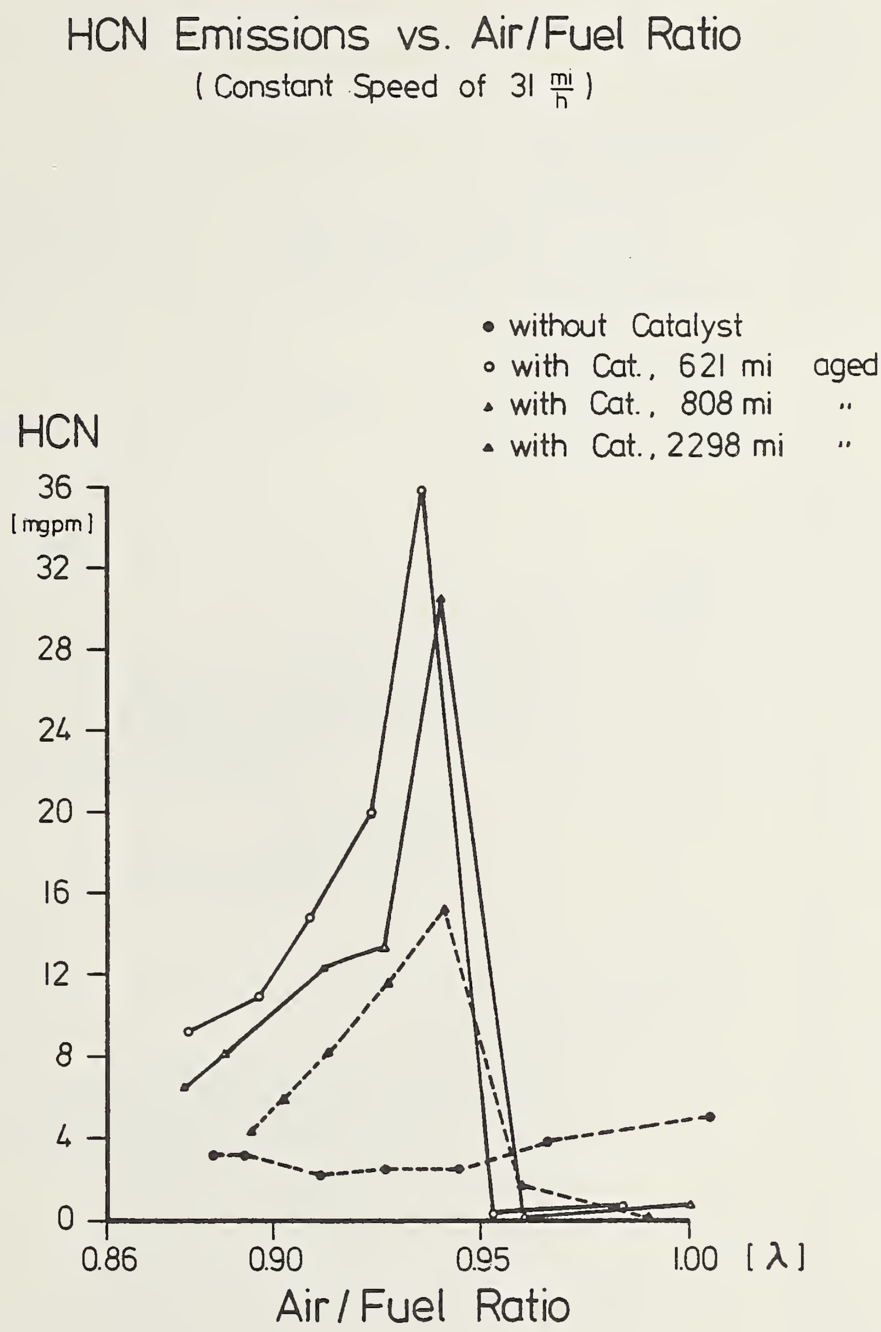

HCN Emissions vs. Air/Fuel Ratio
(Constant Speed of 31 mi/h)
• without Catalyst
○ with Cat., 621 mi aged
▴ with Cat., 808 mi ''
▴ with Cat., 2298 mi ''
HCN
[mgpm]
36
32
28
24
20
16
12
8
4
0
0.86
0.90
0.95
1.00
[λ]
Air / Fuel Ratio

Fig.: 6. 2. 27 [11]

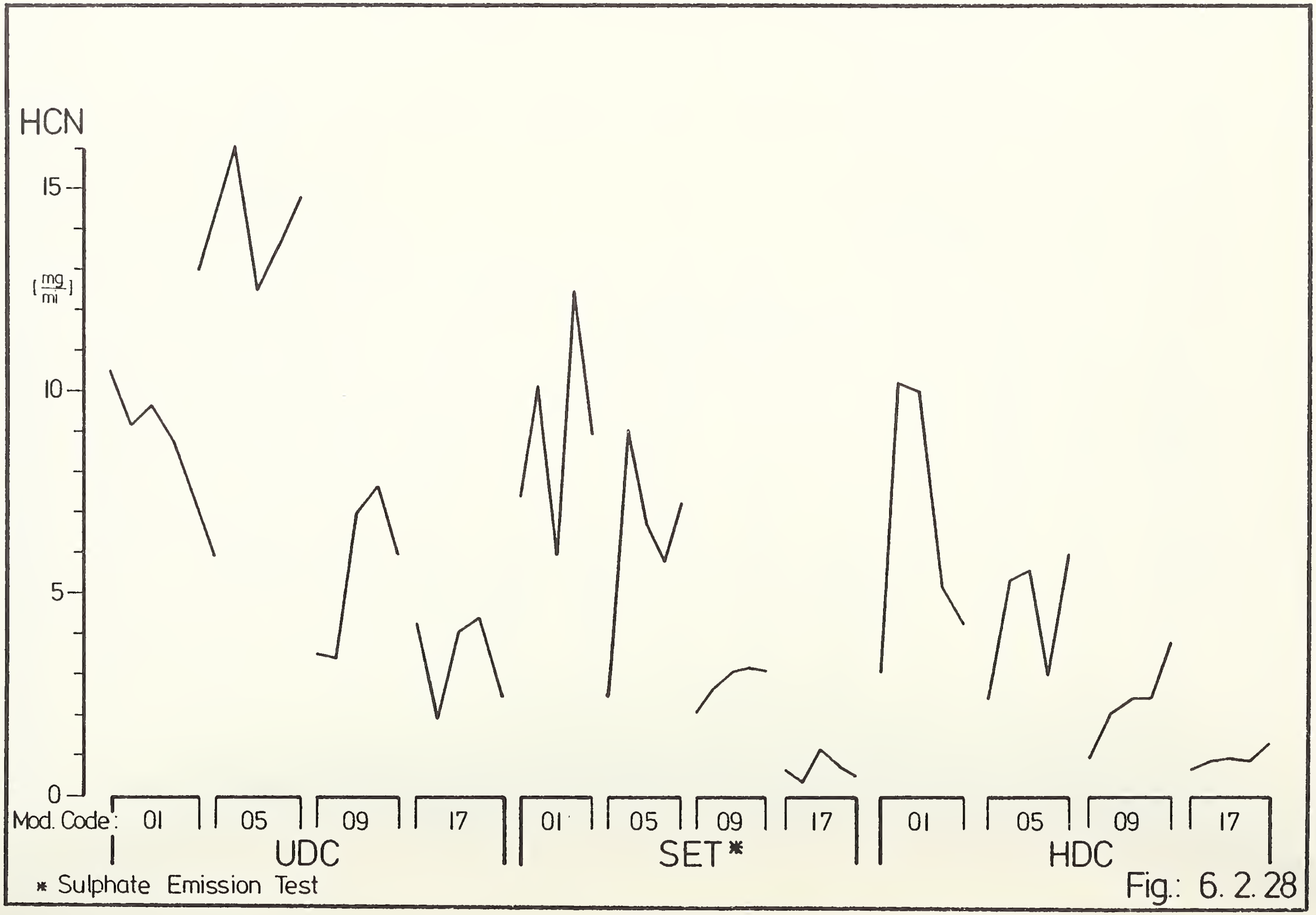
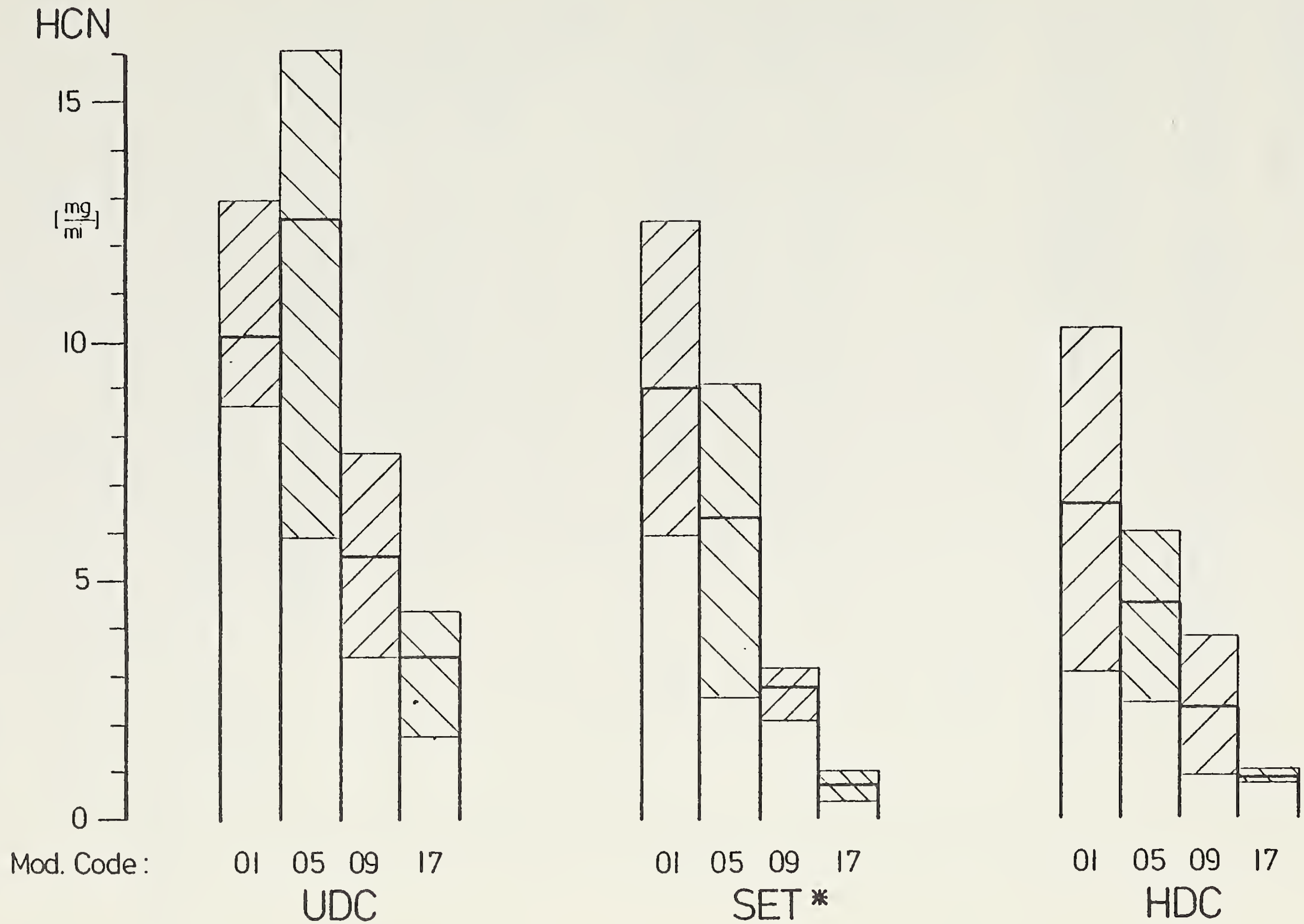

HCN
15
[mg/mi]
10
5
0
Mod. Code: 01 05 09 17 01 05 09 17 01 05 09 17
UDC
SET*
HDC
* Sulphate Emission Test

Fig.: 6.2.28



* Sulphate Emission Test

Fig.: 6. 2. 29

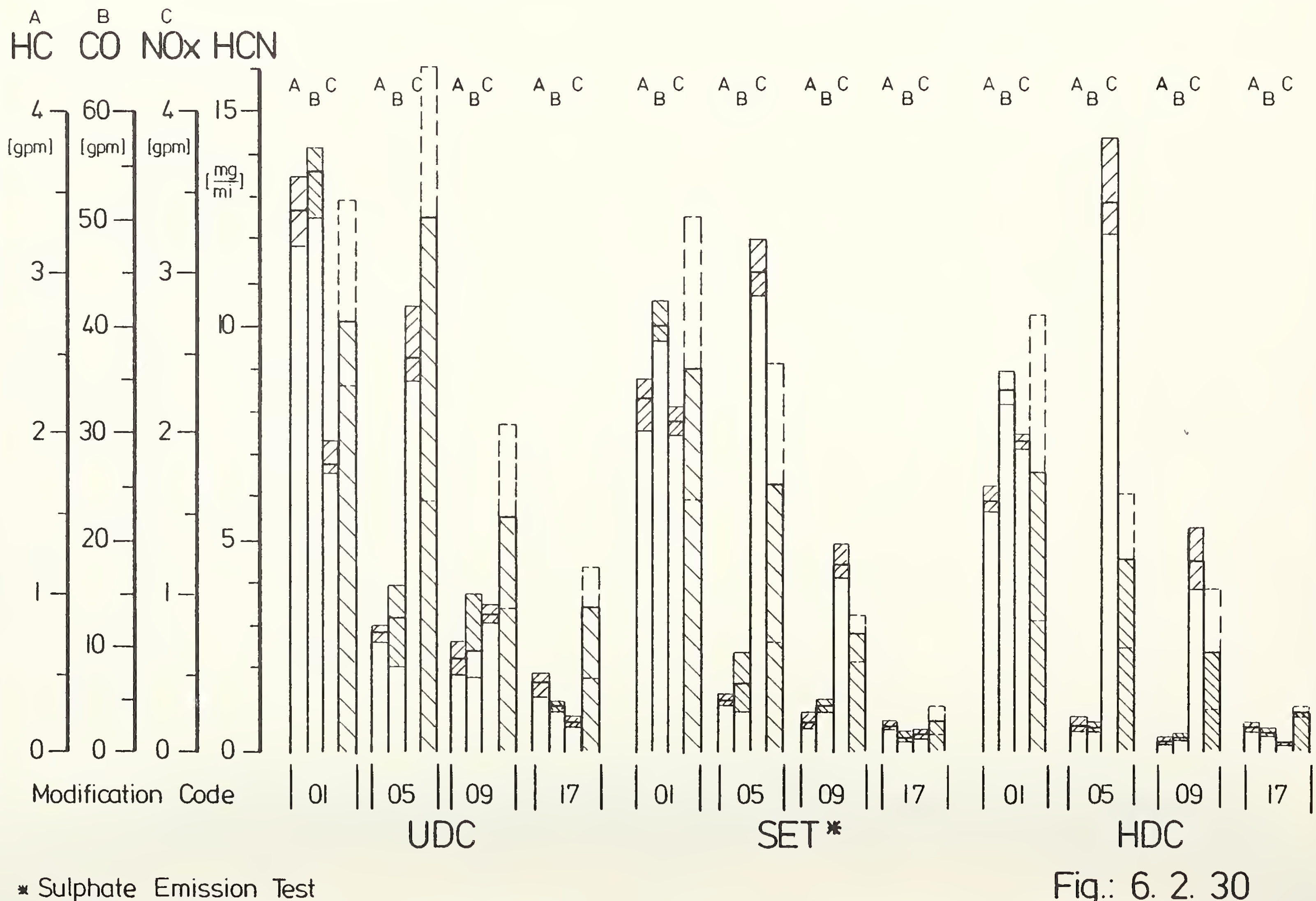
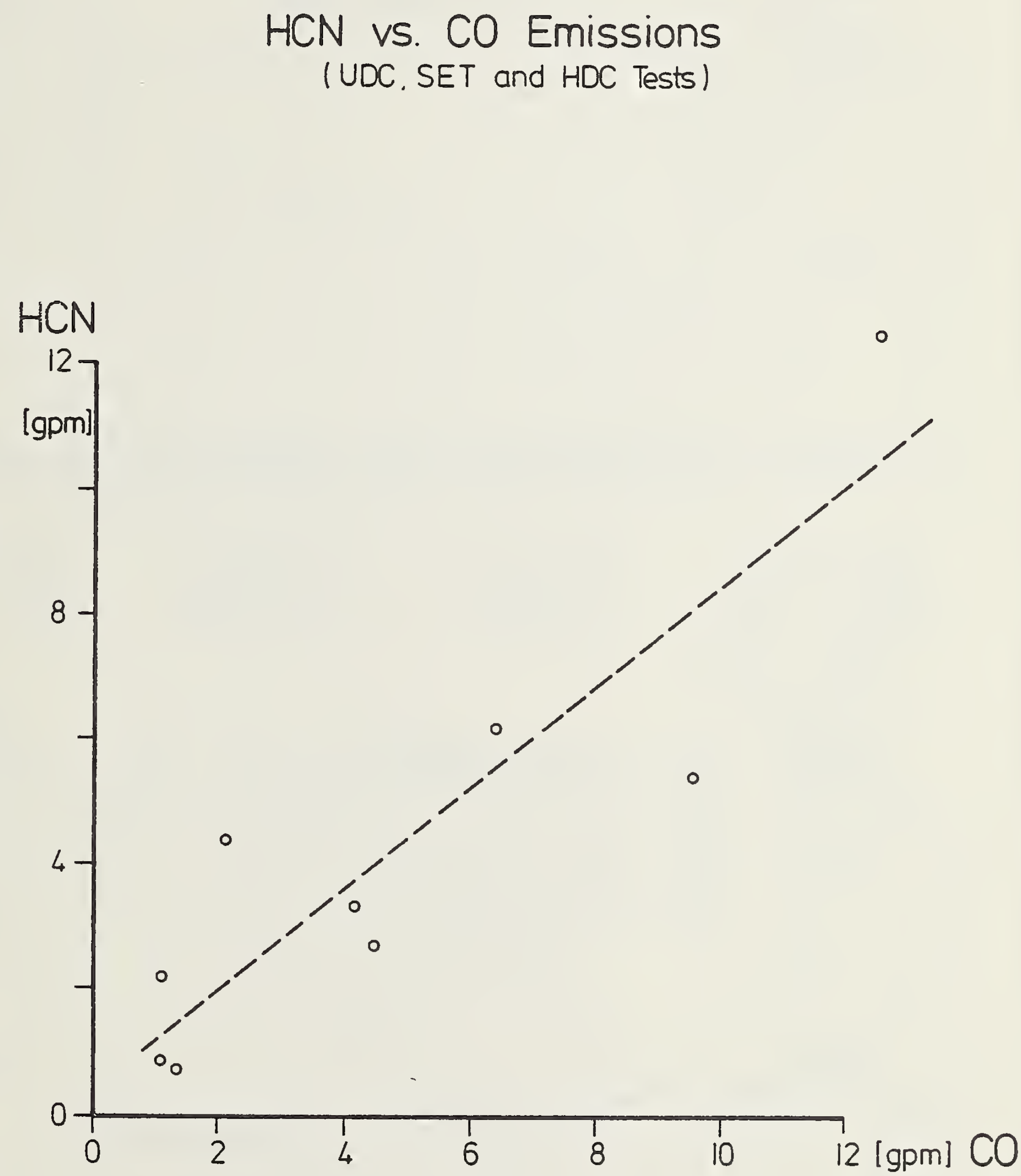

A B C
HC CO NOx HCN
[gpm] [gpm] [gpm] [mg/mi]
Modification Code
01 05 09 17
UDC
SET*
HDC
* Sulphate Emission Test

Fig.: 6. 2. 30



HCN vs. CO Emissions
(UDC, SET and HDC Tests)
HCN
[gpm]
12
8
4
0
0
2
4
6
8
10
12 [gpm] CO

Fig.: 6. 2. 31

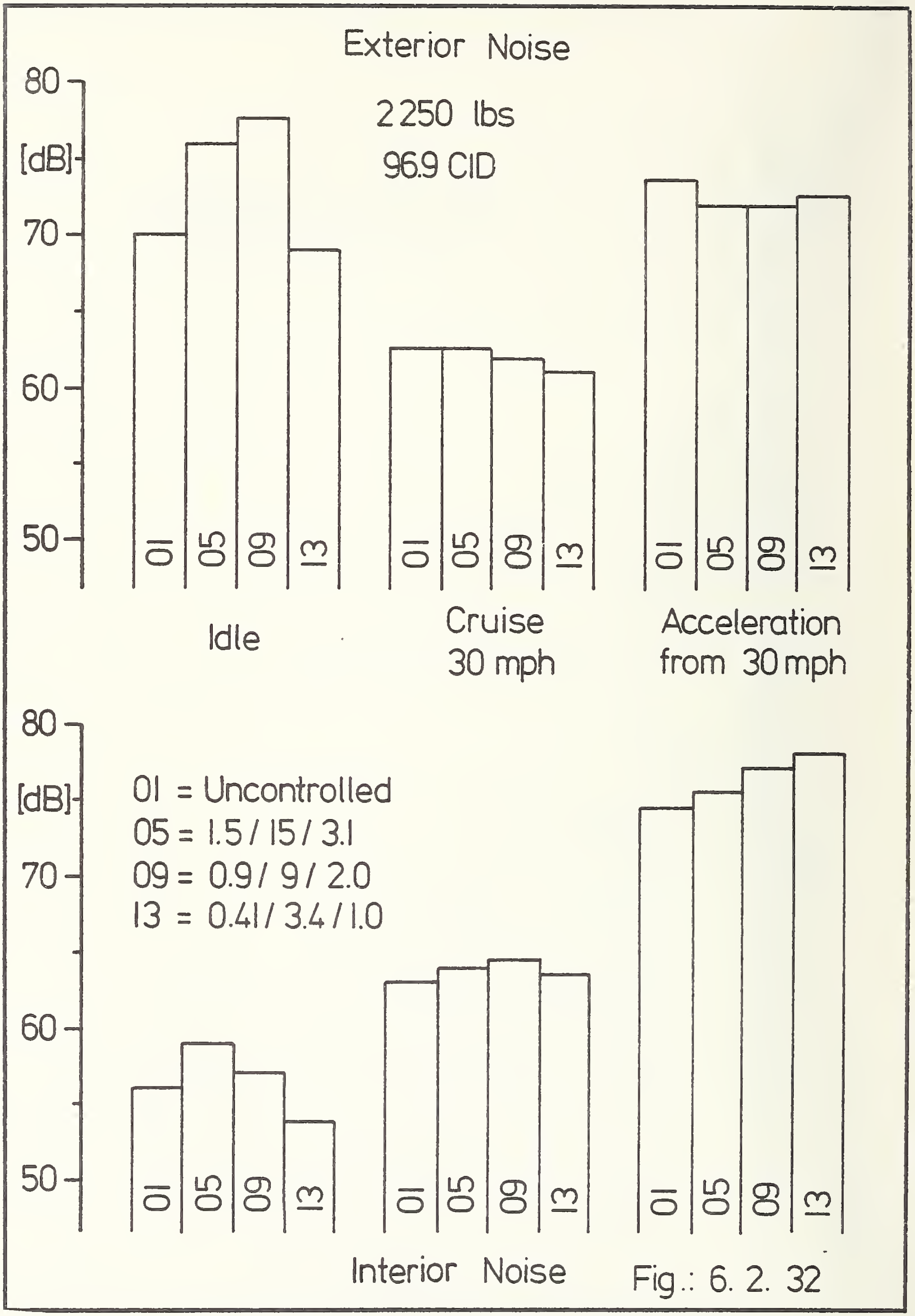

Exterior Noise
2250 lbs
96.9 CID
80
[dB]
70
60
50
01
05
09
13
Idle
Cruise
30 mph
Acceleration
from 30 mph
01 = Uncontrolled
05 = 1.5 / 15 / 3.1
09 = 0.9 / 9 / 2.0
13 = 0.41 / 3.4 / 1.0
Interior Noise

Fig.: 6. 2. 32

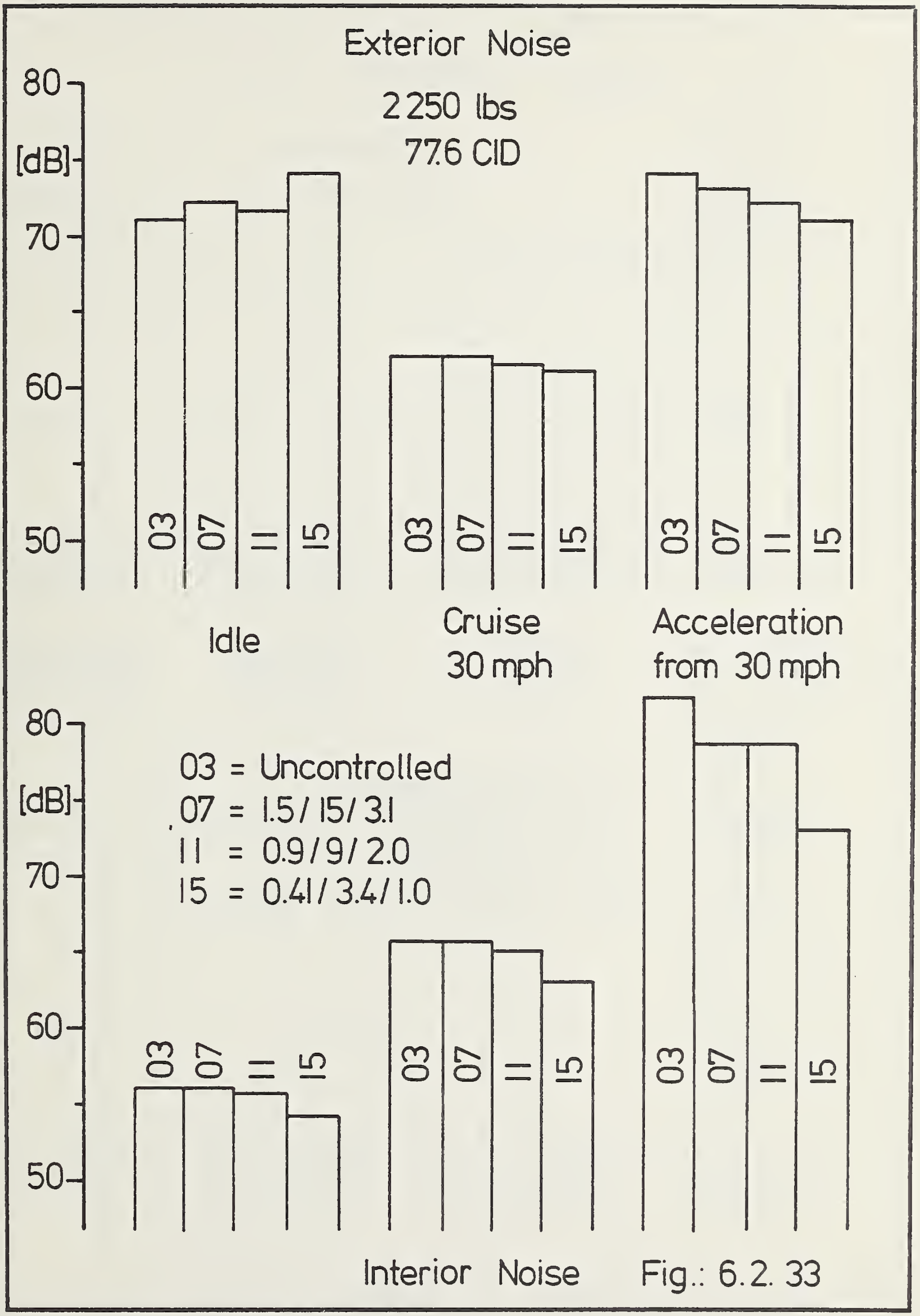

Exterior Noise
2 250 lbs
77.6 CID
80
[dB]
70
60
50
03
07
11
15
Idle
Cruise
30 mph
Acceleration
from 30 mph
03 = Uncontrolled
07 = 1.5 / 15 / 3.1
11 = 0.9 / 9 / 2.0
15 = 0.41 / 3.4 / 1.0
Interior Noise

Fig.: 6.2. 33

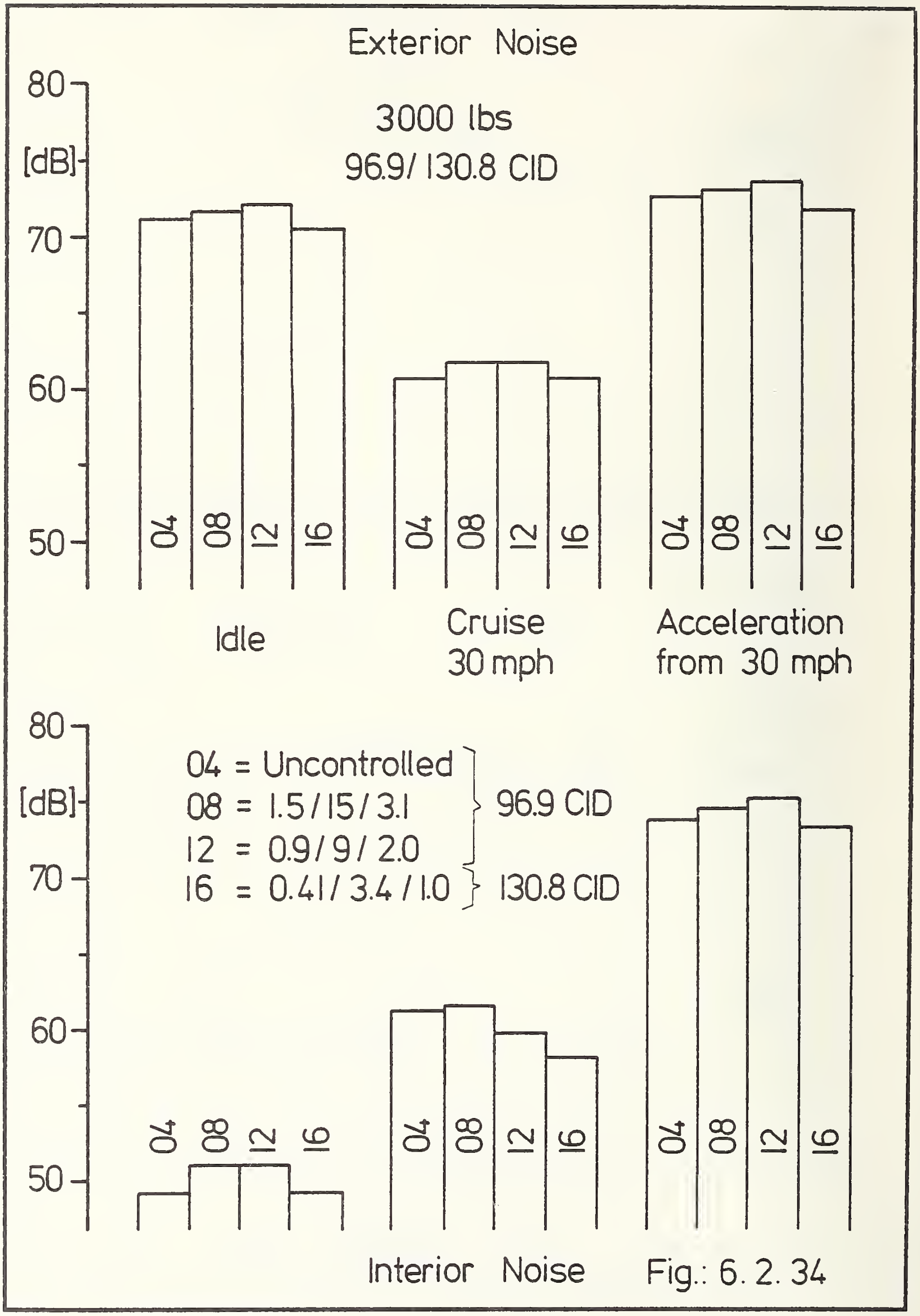

Exterior Noise
80
[dB]
70
60
50
3000 lbs
96.9/ 130.8 CID
04
08
12
16
Idle
Cruise
30 mph
Acceleration
from 30 mph
80
[dB]
70
60
50
04 = Uncontrolled
08 = 1.5 / 15 / 3.1
12 = 0.9 / 9 / 2.0
16 = 0.41 / 3.4 / 1.0
96.9 CID
130.8 CID
Interior Noise

Fig.: 6. 2. 34

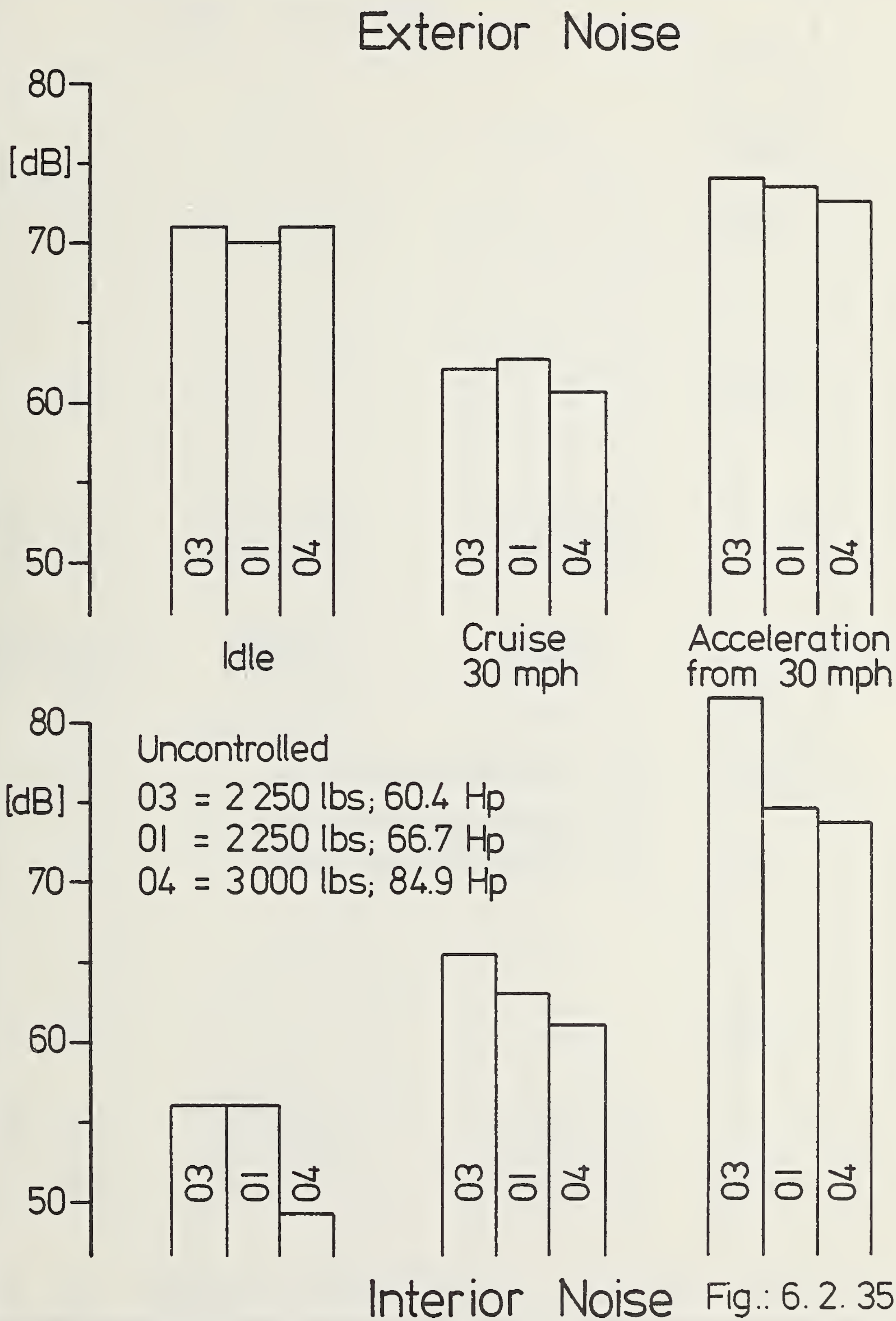

Exterior Noise
80
[dB]
70
60
50
03
01
04
Idle
Cruise
30 mph
Acceleration
from 30 mph
Uncontrolled
03 = 2 250 lbs; 60.4 Hp
01 = 2 250 lbs; 66.7 Hp
04 = 3 000 lbs; 84.9 Hp
Interior Noise

Fig.: 6. 2. 35

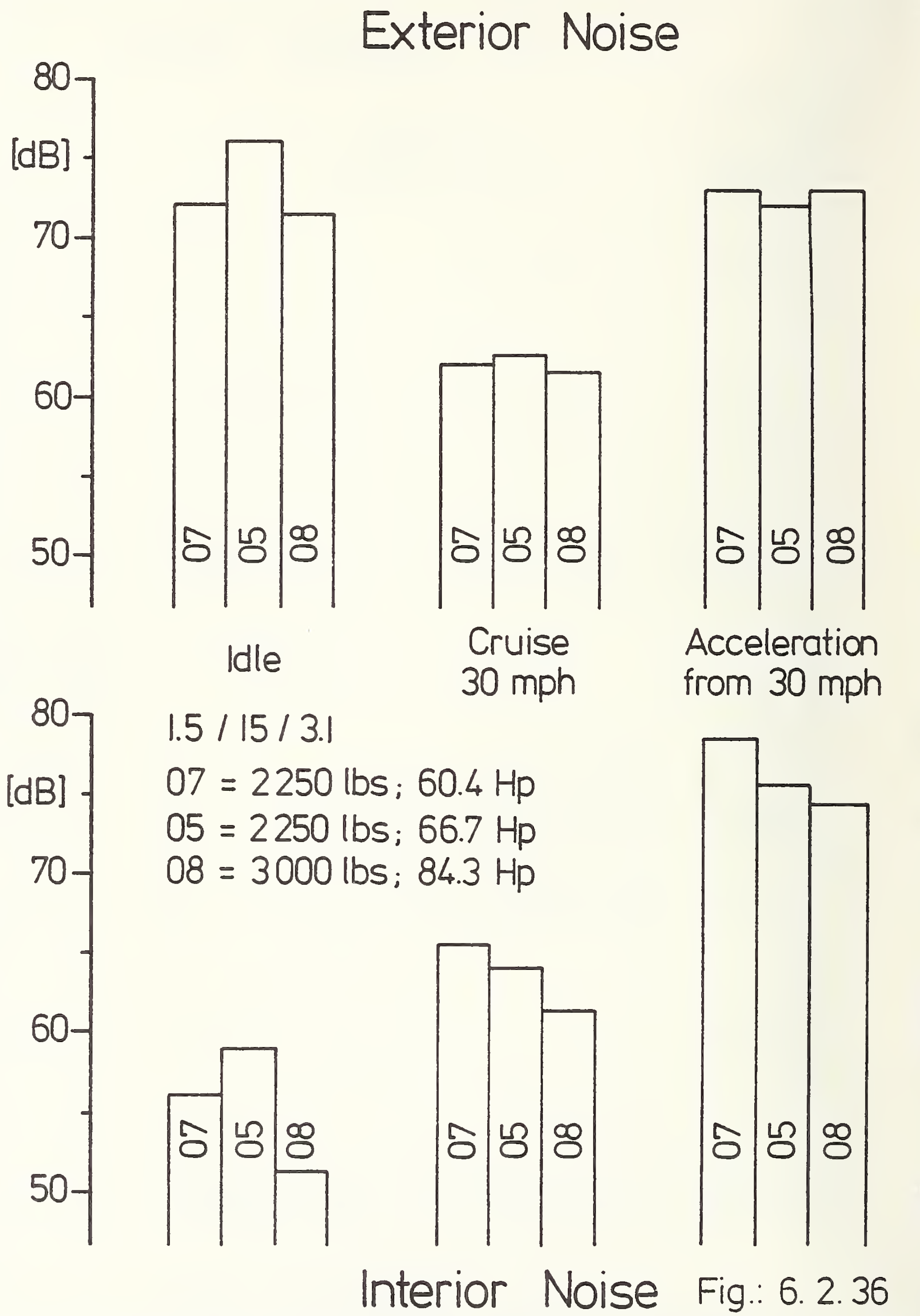
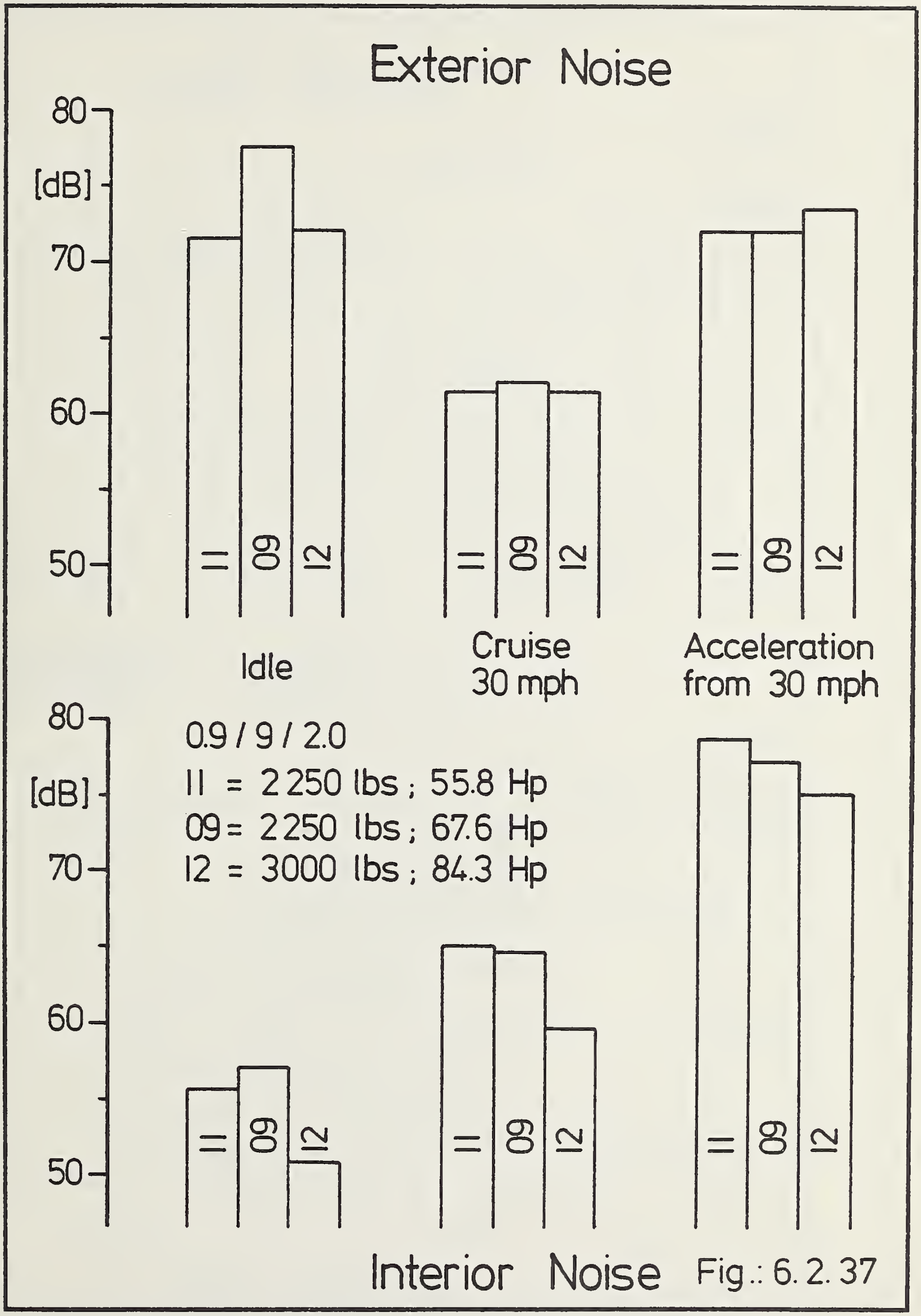

Exterior Noise
80
[dB]
70
60
50
07
05
08
07
05
08
07
05
08
Idle
Cruise
30 mph
Acceleration
from 30 mph
80
[dB]
70
60
50
1.5 / 15 / 3.1
07 = 2250 lbs; 60.4 Hp
05 = 2250 lbs; 66.7 Hp
08 = 3000 lbs; 84.3 Hp
07
05
08
07
05
08
07
05
08
Interior Noise

Fig.: 6. 2. 36


Exterior Noise
80
[dB]
70
60
50
11
09
12
Idle
Cruise
30 mph
Acceleration
from 30 mph
0.9 / 9 / 2.0
11 = 2 250 lbs ; 55.8 Hp
09 = 2 250 lbs ; 67.6 Hp
12 = 3000 lbs ; 84.3 Hp
Interior Noise
Fig.: 6. 2. 37

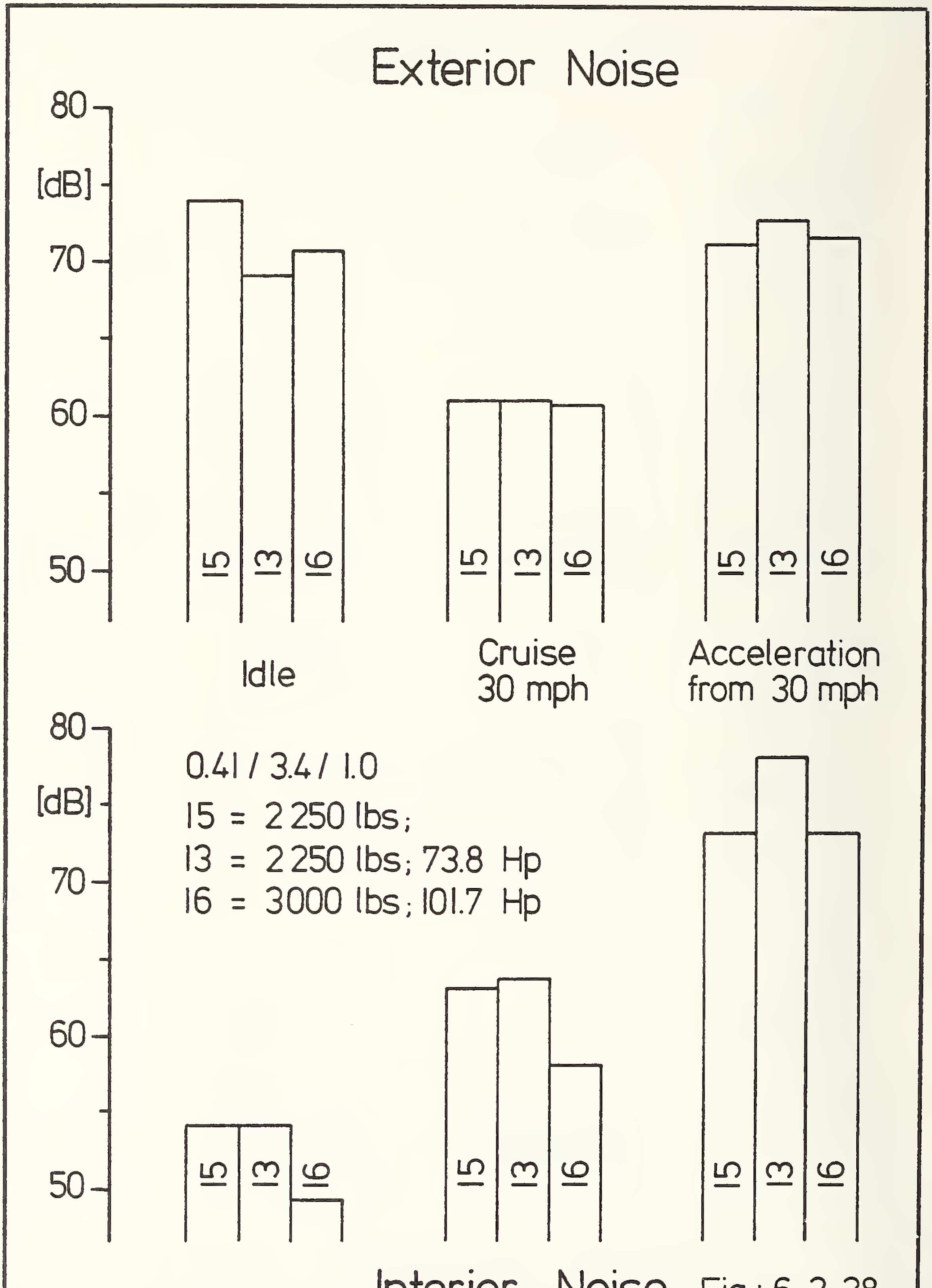

Exterior Noise
80
[dB]
70
60
50
15
13
16
Idle
Cruise
30 mph
Acceleration
from 30 mph
0.41 / 3.4 / 1.0
15 = 2 250 lbs;
13 = 2 250 lbs; 73.8 Hp
16 = 3000 lbs; 101.7 Hp
Interior Noise

Fig.: 6. 2. 38

Vibration Pick up Points

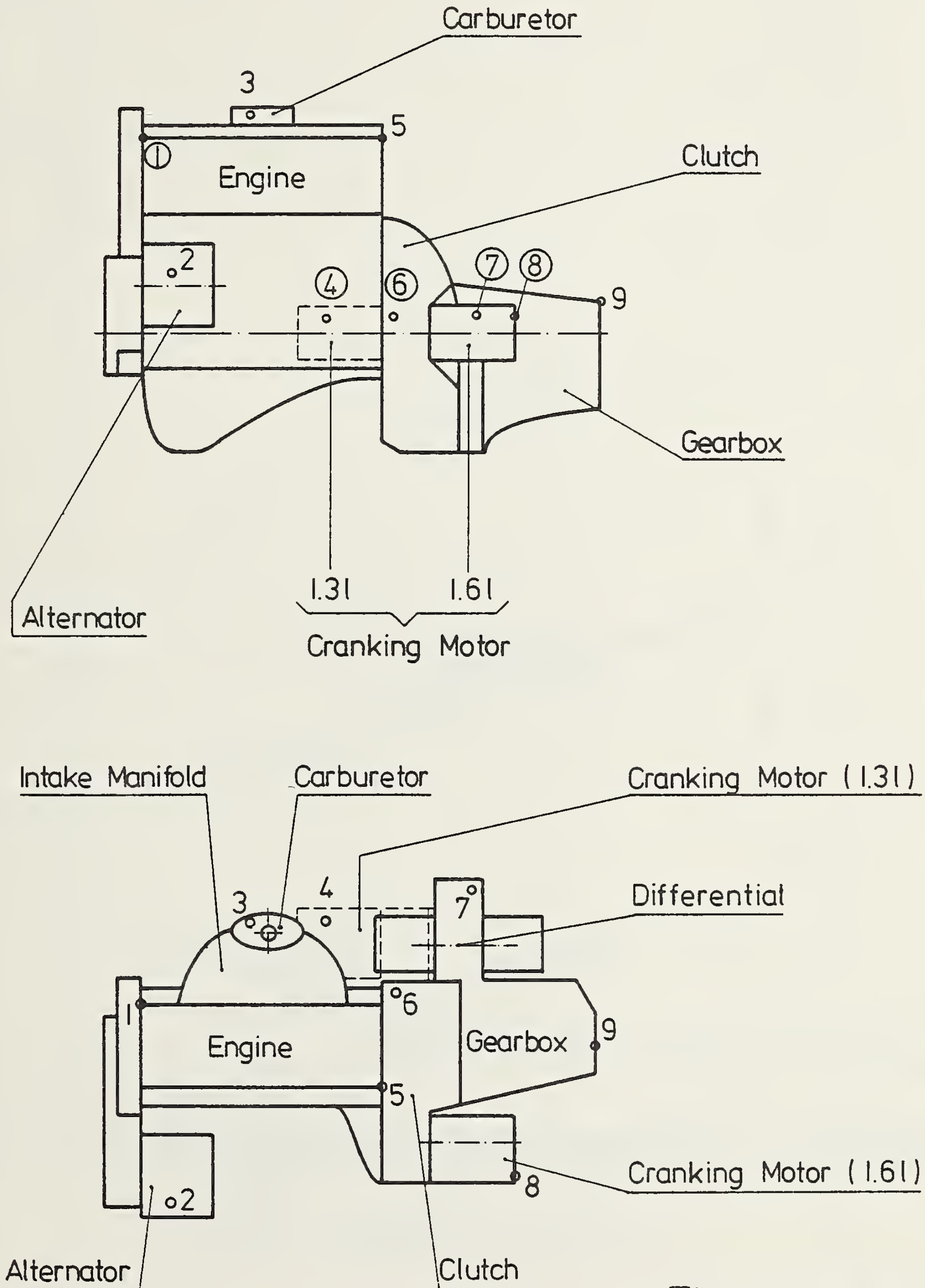


Fig.: 6.2. 39

O Encircled Pick up Points Behind the Engine

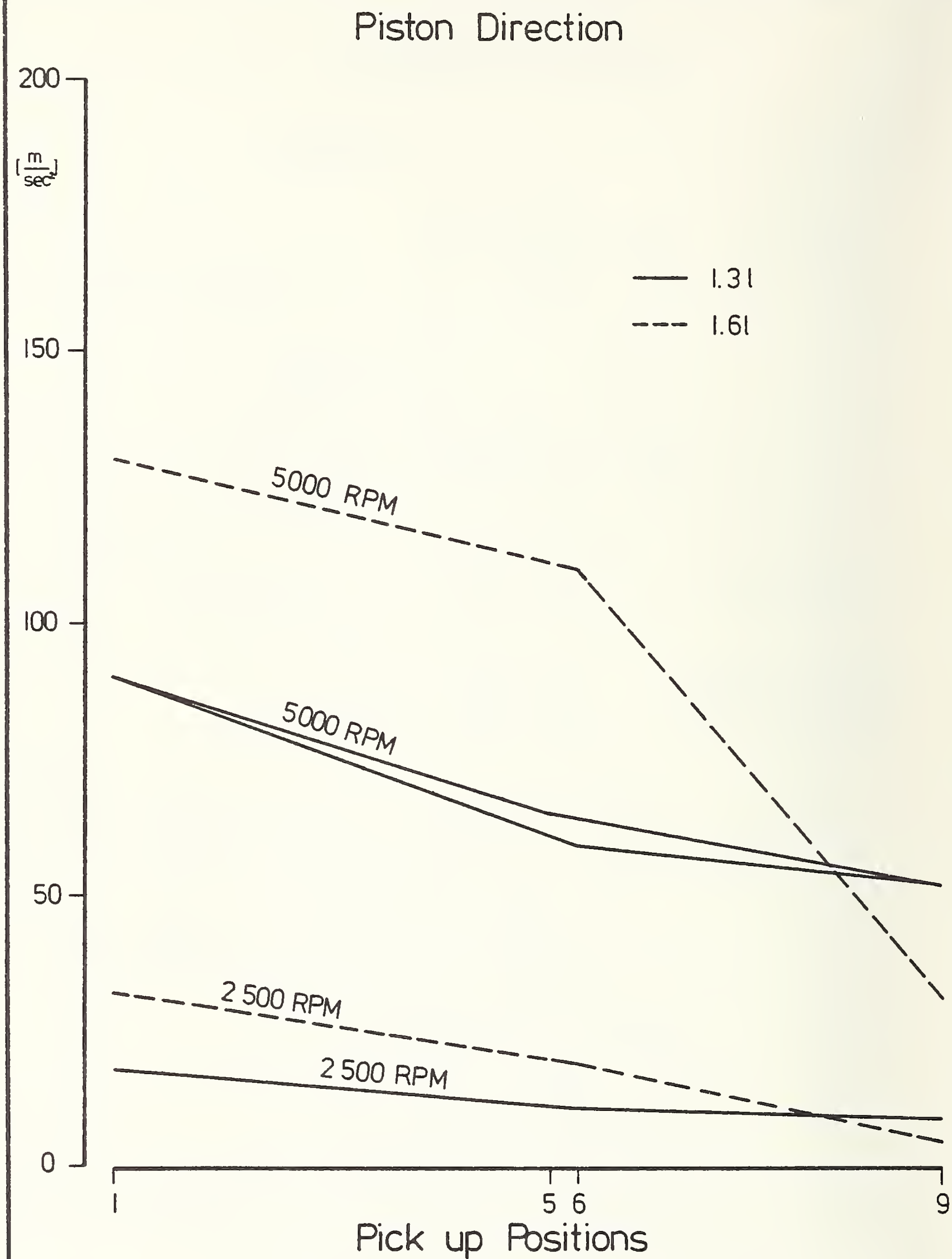

Piston Direction
200
150
100
50
0
[$\frac{m}{sec^2}$]
1.3 l
1.6 l
5000 RPM
5000 RPM
2 500 RPM
2 500 RPM
1
5
6
9
Pick up Positions

Fig.: 6. 2. 40

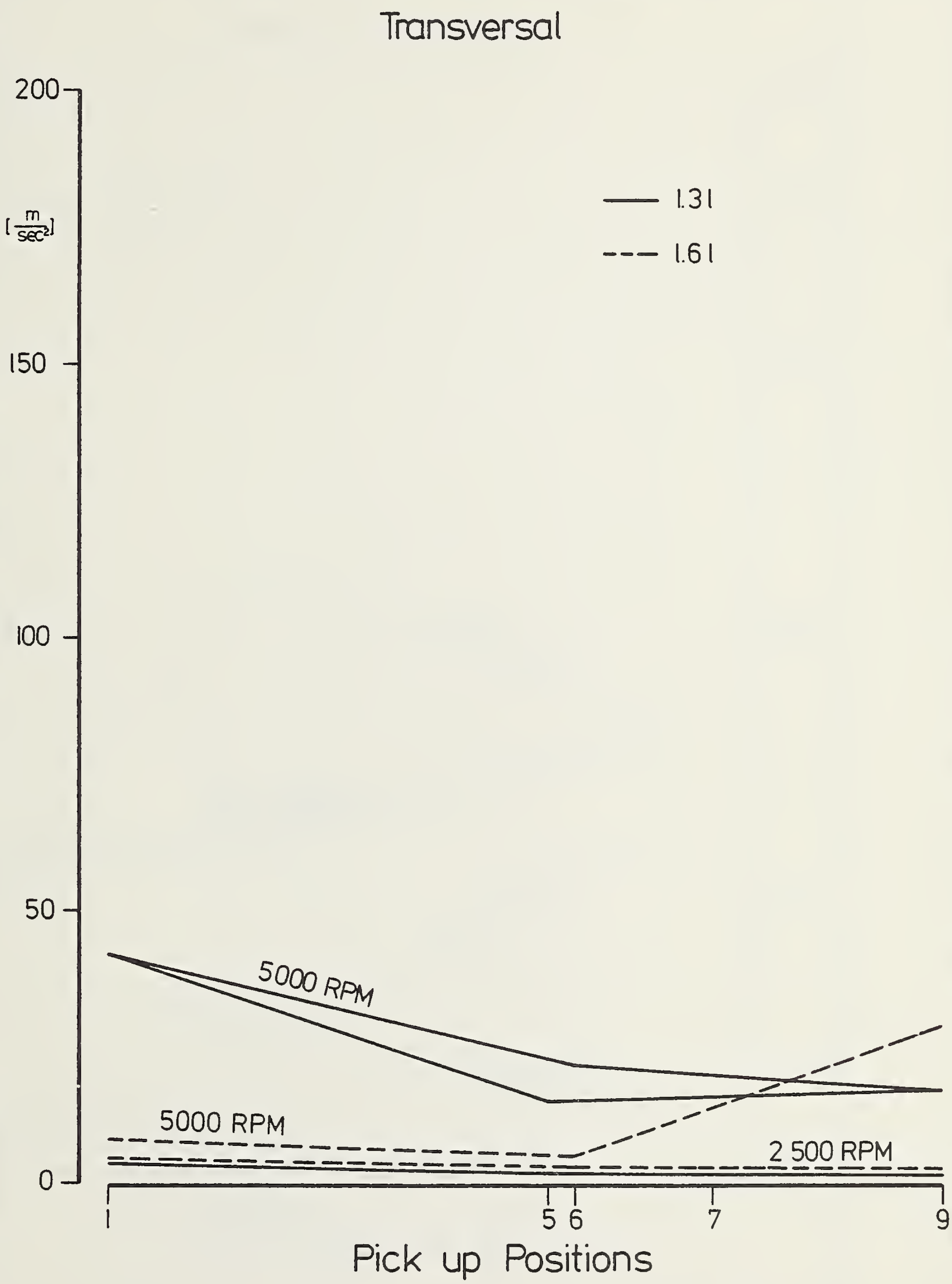

Transversal
200
150
100
50
0
[m/sec²]
1.3 l
1.6 l
5000 RPM
5000 RPM
2 500 RPM
1
5 6
7
9
Pick up Positions

Fig.: 6.2.41

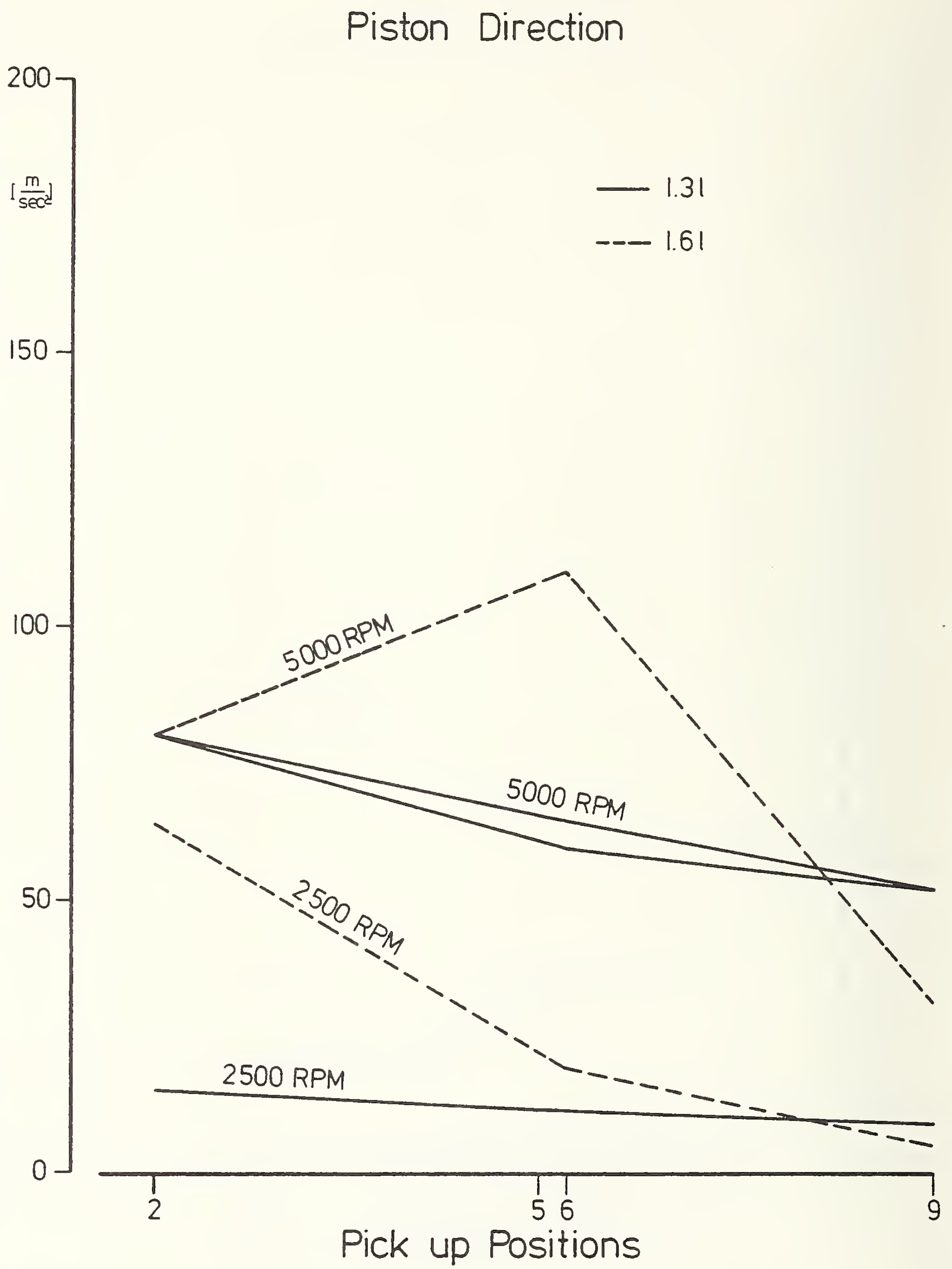
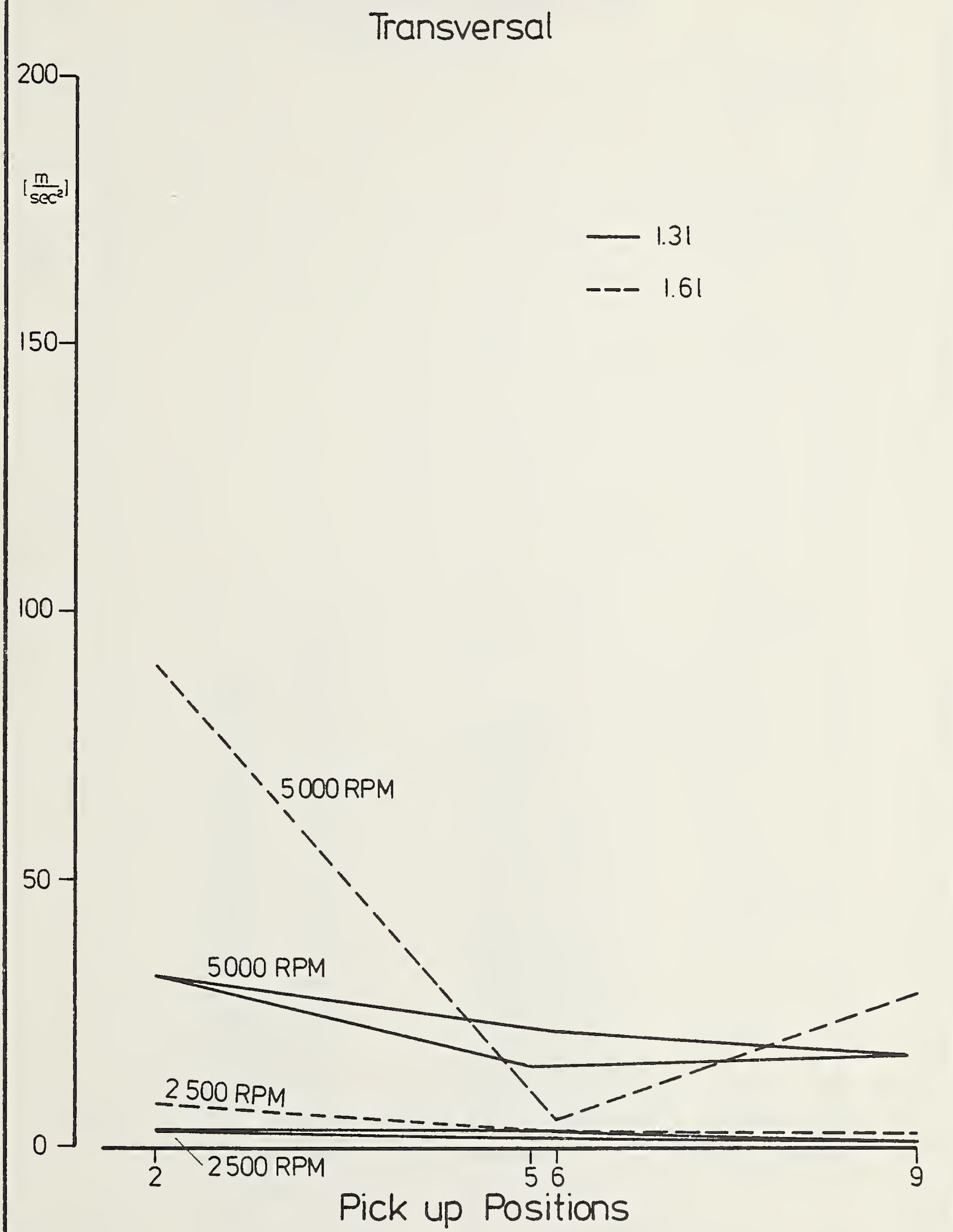
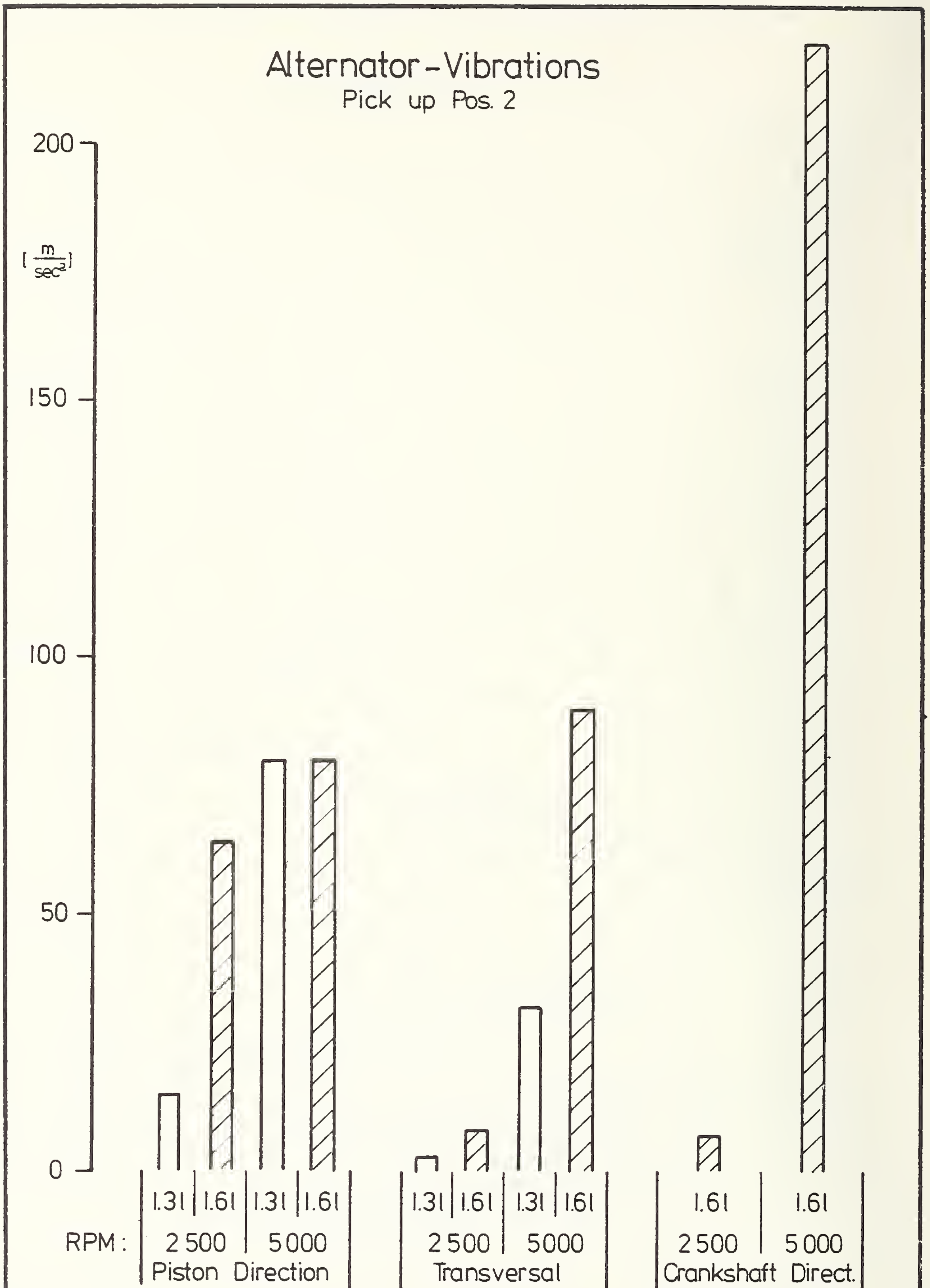

Piston Direction
200
150
100
50
0
[m/sec²]
1.31
1.61
5000 RPM
5000 RPM
2500 RPM
2500 RPM
2
5
6
9
Pick up Positions

Fig.: 6.2.42



Transversal
200
$[\frac{m}{sec^2}]$
150
100
50
0
1.31
1.61
5 000 RPM
5000 RPM
2 500 RPM
2 500 RPM
2
5 6
9
Pick up Positions

Fig.: 6. 2. 43



Alternator - Vibrations
Pick up Pos. 2
200
[m/sec²]
150
100
50
0
1.31 1.61 1.31 1.61
RPM : 2 500 5 000
Piston Direction
1.31 1.61 1.31 1.61
2 500 5000
Transversal
1.61 1.61
2 500 5 000
Crankshaft Direct.

Fig.: 6. 2.44

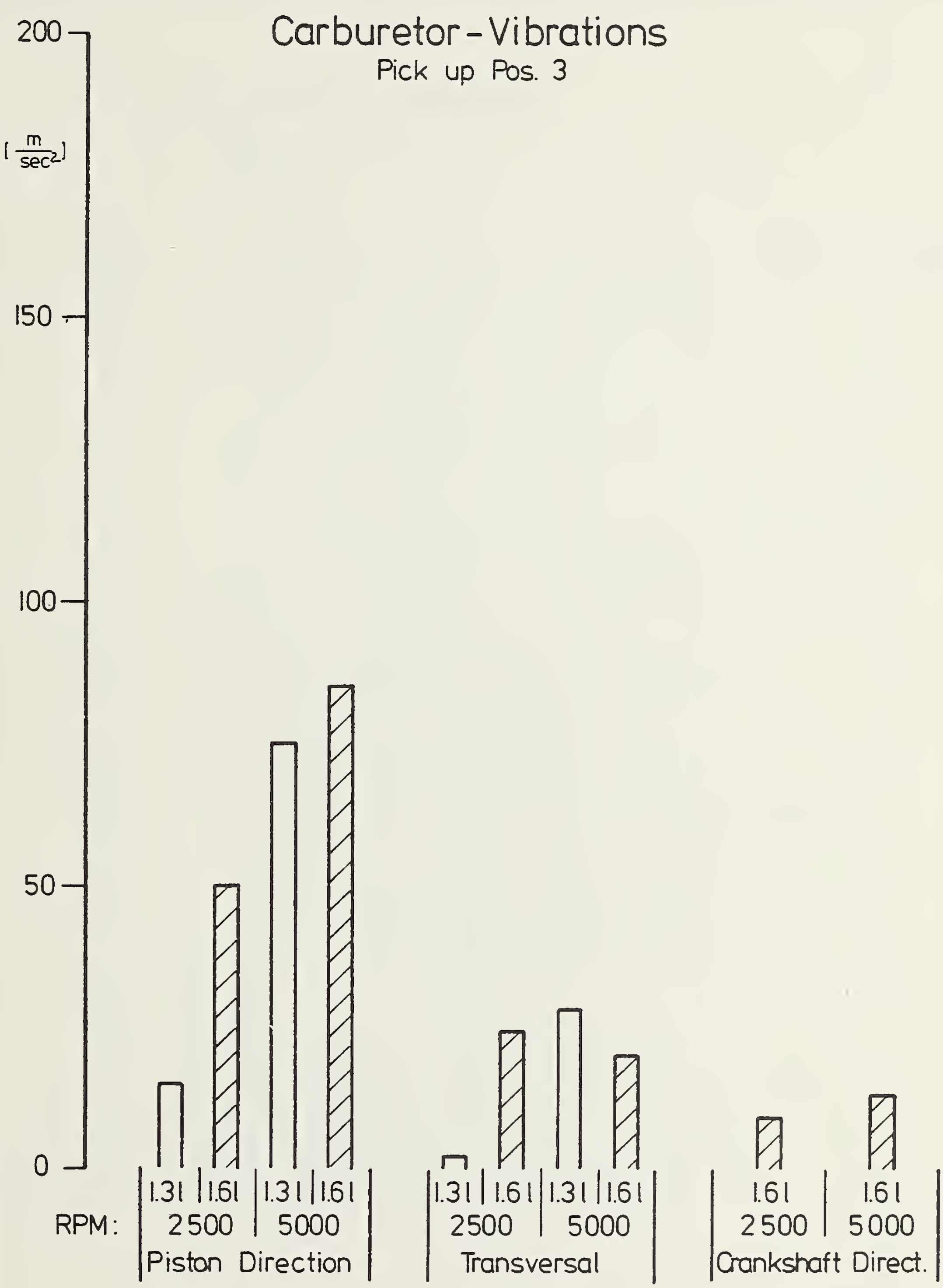

Carburetor - Vibrations
Pick up Pos. 3
200
[m/sec²]
150
100
50
0
RPM:
1.3 l
1.6 l
1.3 l
1.6 l
2500
5000
Piston Direction
1.3 l
1.6 l
1.3 l
1.6 l
2500
5000
Transversal
1.6 l
1.6 l
2500
5000
Crankshaft Direct.

Fig.: 6. 2. 45

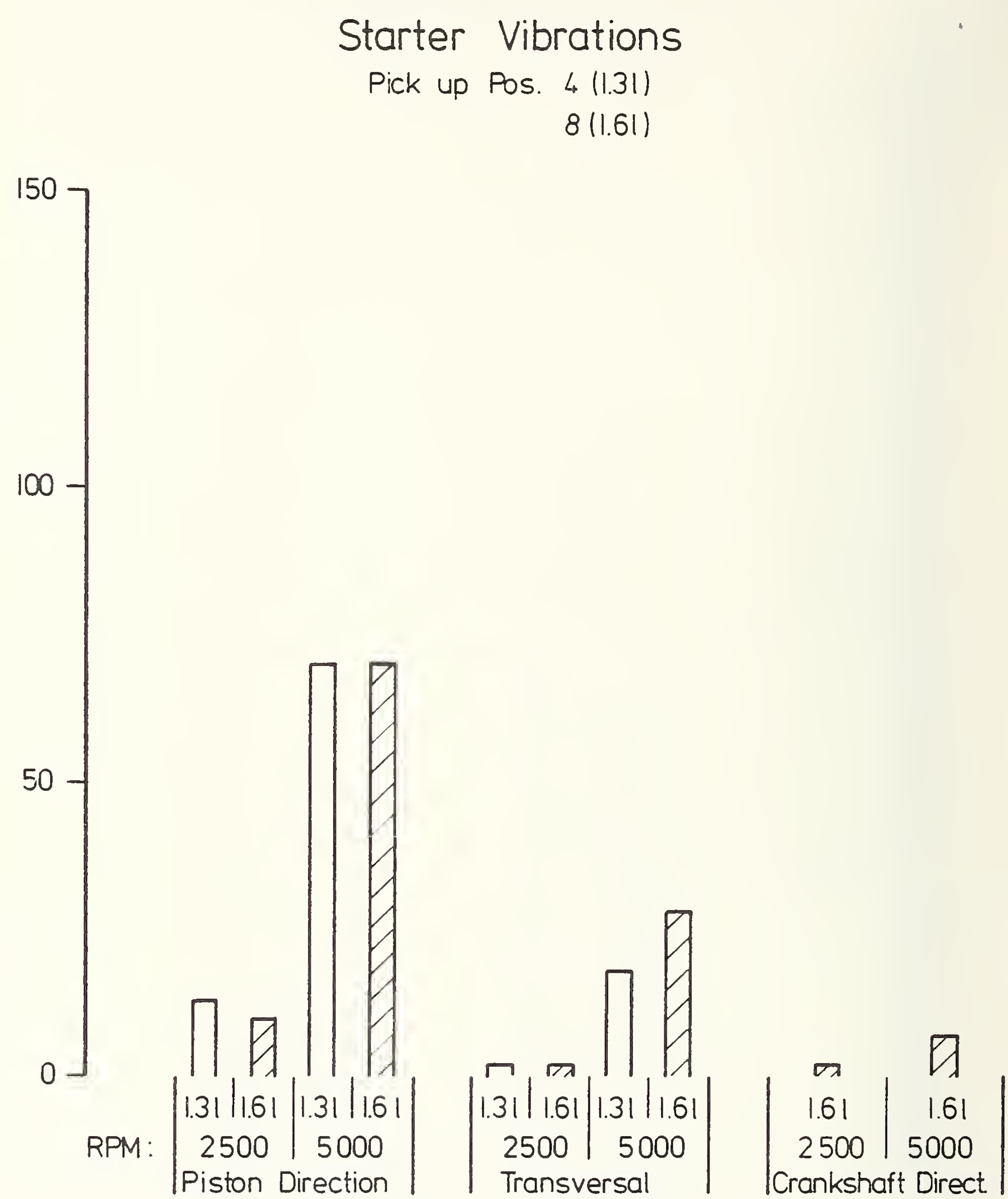

Starter Vibrations
Pick up Pos. 4 (1.31)
8 (1.61)
150
100
50
0
1.31
1.61
1.31
1.61
RPM:
2500
5000
Piston Direction
1.31
1.61
1.31
1.61
2500
5000
Transversal
1.61
1.61
2500
5000
Crankshaft Direct.

Fig.: 6. 2. 46

Reproducibility on Engine and Chassis Dynomometers

Engine / Vehicle	Chassis Dynomometer			Engine Dynomometer		
	HC [gpm]	CO [gpm]	NOx [gpm]	HC [gpm]	CO [gpm]	NOx [gpm]
Mod. Code 17	0.35	4.12	0.15	0.16	1.09	0.11
"	0.45	4.07	0.19	0.13	0.82	0.12
"	0.46	4.62	0.18	0.12	0.95	0.13
"	0.49	4.14	0.18	0.11	0.74	0.11
"	0.49	4.25	0.14	0.14	0.75	0.13

Tab.: 6. 2.1

Description of vehicle	
Engine family	2
Model	Convertible 15
Engine Code	AJ
Displacement	97 cu.in.
Number of cylinders	4
Compression ratio	7.3
Advertised HP	48 SAE net
Bore	3.37 in
Stroke	2.72 in
Transmission	M 4
Axle ratio	3.875
N/V	49.9
Tire size	6.00-15 L
EFI make	Bosch
Exhaust control system	EFI/EGR/CAT
Evap control system	carbon canister
Crankcase control system	closed system
Idle speed	875 ± 75 RPM
Idle CO	0.6 ± 0.4
Ignition timing	5 ATDC
Dwell angle	47 ± 3
Inertia weight	2,500
Catalyst	Oxidation catalyst Degussa OM 722/14L

Tab.: 6.2.2

Vehicle No.	HC gpm	CO gpm	NO_x gpm	CH_4 mgpm	Percent Methane from THC
1	0.40	7.20	0.94	83.3	20.8
2	0.41	6.67	1.08	87.7	21.4
3	0.36	5.21	1.21	78.0	21.7
4	0.45	4.66	0.86	95.7	21.3
5	0.37	3.40	1.14	77.1	20.8
6	0.42	4.86	0.91	97.3	23.2
7	0.39	4.96	0.79	95.8	24.6
8	0.37	5.05	0.93	85.1	23.0
9	0.41	4.05	0.92	83.6	20.4
10	0.38	5.50	0.99	92.6	24.4
$\bar{x}$ average	0.40	5.2	0.98	87.6	22.2
s standard deviation	0.03	1.1	0.13	7.4	1.5

Tab.: 6. 2. 3

Mod. Code	Emission Tasks HC	Emission Tasks CO	Emission Tasks NOx	Inertia Weight 2 250	Inertia Weight 3 000	CVS-Emissions HC Test	HC AV.	CO Test	CO AV.	NOx Test	NOx AV.
01	uncontrolled			×		3.18		15.8		3.29	
01		"		×		3.21		21.7		3.67	
01		"		×		2.58	2.91	26.6	23.8	3.32	3.44
01		"		×		2.80		21.7		3.52	
01		"		×		2.76		33.3		3.41	
02	uncontrolled				×	2.84		24.1		4.39	
02		"			×	2.99		30.9		4.40	
02		"			×	3.14	3.10	28.7	29.3	4.34	4.34
02		"			×	3.39		25.8		4.33	
02		"			×	3.15		37.2		4.26	
03	uncontrolled			×		2.12		14.9		1.97	
03		"		×		2.24		13.7		202	
03		"		×		2.12	2.13	13.0	13.3	2.06	2.01
03		"		×		1.98		12.5		1.92	
03		"		×		2.18		12.6		2.10	
04	uncontrolled				×	3.75		24.5		3.65	
04		"			×	3.08		27.2		3.48	
04		"			×	3.17	3.13	30.8	27.9	3.94	3.74
04		"			×	3.09		27.2		3.88	
04		"			×	2.57		29.9		3.74	

Mod. Code	Emission Tasks HC	Emission Tasks CO	Emission Tasks NOx	Inertia Weight 2 250	Inertia Weight 3 000	CVS-Emissions HC Test	HC AV	CO Test	CO AV.	NOx Test	NOx AV
05	1.5	15	3.1	×		0.54		8.54		2.19	
05	"	"	"	×		0.63		9.20		1.96	
05	"	"	"	×		0.57	0.59	5.29	7.26	2.01	2.18
05	"	"	"	×		0.58		6.60		2.40	
05	"	"	"	×		0.64		6.66		2.34	
06	1.5	15	3.1		×	1.07		12.7		1.88	
06	"	"	"		×	0.60		7.88		2.12	
06	"	"	"		×	0.60	0.71	8.72	9.69	1.95	2.00
06	"	"	"		×	0.65		8.80		1.97	
06	"	"	"		×	0.63		10.4		206	
07	1.5	15	3.1	×		0.75		6.79		1.50	
07	"	"	"	×		0.93		7.44		2.31	
07	"	"	"	×		0.97	0.95	8.76	8.08	2.40	2.32
07	"	"	"	×		1.31		10.1		2.52	
07	"	"	"	×		0.80		7.38		2.86	
08	1.5	15	3.1		×	1.17		10.5		1.85	
08	"	"	"		×	1.37		10.0		1.72	
08	"	"	"		×	1.43	1.26	11.3	10.6	1.79	1.77
08	"	"	"		×	1.15		10.7		1.80	
08	"	"	"		×	1.17		10.3		1.67	

Mod. Code	Emission Tasks HC	Emission Tasks CO	Emission Tasks NOx	Inertia Weight 2 250	Inertia Weight 3 000	CVS-Emissions HC Test	HC AV.	CO Test	CO AV.	NOx Test	NOx AV
09	0.9	9.0	2.0	×		0.38		3.57		1.54	
09	"	"	"	×		0.32		3.90		1.35	
09	"	"	"	×		0.37	0.34	5.38	4.03	1.19	1.40
09	"	"	"	×		0.32		3.95		1.37	
09	"	"	"	×		0.32		3.35		1.53	
10	09	90	2.0		×	0.46		6.92		1.82	
10	"	"	"		×	0.35		5.95		1.76	
10	"	"	"		×	0.43	0.40	7.90	6.50	1.75	1.76
10	"	"	"		×	0.39		6.86		1.52	
10	"	"	"		×	0.38		4.85		1.94	
11	0.9	9.0	2.0	×		0.66		5.91		1.93	
11	"	"	"	×		0.80		7.70		1.97	
11	"	"	"	×		0.63	0.72	6.25	6.67	1.20	1.52
11	"	"	"	×		0.79		7.51		1.21	
11	"	"	"	×		0.71		6.00		1.28	
12	0.9	9.0	2.0		×	0.62		7.94		1.74	
12	"	"	"		×	0.58		6.12		1.80	
12	"	"	"		×	0.63	0.62	8.89	7.63	1.72	1.74
12	"	"	"		×	0.63		7.98		1.70	
12	"	"	"		×	0.65		7.21		1.74	

Tab.: 6.2.4

Mod. Code	Emission Tasks			Inertia Weight		CVS-Emissions					
						HC		CO		NOx	
	HC	CO	NOx	2 250	3 000	Test	AV.	Test	AV.	Test	AV.
13	0.41	3.4	1.0	×		0.15		0.62		0.21	
13	"	"	"	×		0.12		1.00		0.18	
13	"	"	"	×		0.11	0.12	0.64	0.81	0.19	0.18
13	"	"	"	×		0.12		0.85		0.21	
13	"	"	"	×		0.09		0.94		0.10	
14	0.41	3.4	1.0		×	0.14		1.23		0.15	
14	"	"	"		×	0.19		1.22		0.19	
14	"	"	"		×	0.13	0.16	0.98	1.24	0.17	0.18
14	"	"	"		×	0.18		1.06		0.17	
14	"	"	"		×	0.14		1.70		0.20	
15	0.41	3.4	1.0	×		0.14		1.57		0.17	
15	"	"	"	×		0.16		1.58		0.13	
15	"	"	"	×		0.14	0.15	1.29	1.52	0.13	0.13
15	"	"	"	×		0.17		2.00		0.11	
15	"	"	"	×		0.12		1.16		0.12	
16	0.41	3.4	1.0		×	0.09		1.44		0.18	
16	"	"	"		×	0.10		1.35		0.19	
16	"	"	"		×	0.13	0.10	1.25	1.30	0.18	0.18
16	"	"	"		×	0.06		1.19		0.19	
16	"	"	"		×	0.10		1.25		0.18	

Mod. Code	Emission Tasks			Inertia Weight		CVS-Emissions					
						HC		CO		NOx	
	HC	CO	NOx	2 250	3 000	Test	AV	Test	AV.	Test	AV.
17	0.41	3.4	0.4	×		0.16		1.09		0.11	
17	"	"	"	×		0.13		0.82		0.12	
17	"	"	"	×		0.12	0.13	0.95	0.87	0.13	0.12
17	"	"	"	×		0.11		0.74		0.11	
17	"	"	"	×		0.14		0.75		0.13	
18	0.41	3.4	0.4		×	0.22		1.91		0.10	
18	"	"	"		×	0.22		1.09		0.11	
18	"	"	"		×	0.13	0.16	1.68	1.51	0.14	0.13
18	"	"	"		×	0.10		1.40		0.14	
18	"	"	"		×	0.11		1.46		0.15	
19	0.41	3.4	0.4	×		0.26		1.57		0.09	
19	"	"	"	×		0.22		1.47		0.11	
19	"	"	"	×		0.27	0.23	1.84	1.66	0.10	0.10
19	"	"	"	×		0.21		1.65		0.11	
19	"	"	"	×		0.21		1.78		0.10	
20	0.41	3.4	0.4		×	0.05		1.17		0.14	
20	"	"	"		×	0.09		1.58		0.14	
20	"	"	"		×	0.05	0.06	1.23	1.28	0.15	0.15
20	"	"	"		×	0.05		1.26		0.15	
20	"	"	"		×	0.05		1.17		0.15	

Tab.: 6.2.5

Test-No.	HC [gpm]	CO [gpm]	NOx [gpm]	FE [mpg]	FC [$\frac{gal}{100 mi}$]
18	0.13	1.43	0.20	24.4	4.10
19	0.29	4.74	0.17	25.0	4.00
20	0.24	1.98	0.18	24.9	4.02
21	0.16	1.74	0.22	25.2	3.97
23	0.23	2.64	0.20	24.0	4.17
24	0.23	1.27	0.19	24.4	4.10
25	0.32	0.85	0.21	23.9	4.18
26	0.27	2.15	0.12	23.7	4.22

Tab.: 6. 2. 6

Test-No.	UDC cold	HC [gpm]	CO [gpm]	NOx [gpm]	FE [mpg]	FC [$\frac{gal}{100\,mi}$]	
44	x	0.33	2.25	0.14	23.8	4.20	
46	x	0.23	1.47	0.30	24.2	4.13	
47	x	0.29	1.97	0.10	21.2	4.72	
48	x	0.18	1.62	0.15	24.1	4.15	
49	x	0.37	1.58	0.11	22.4	4.46	
52	x	0.19	1.17	0.15	24.2	4.13	Repetition of Test-No. 48
58	x	0.24	1.86	0.10	22.6	4.42	

Tab.: 6. 2. 7

UDC		HDC	HC	CO	NOx	FE	FC	Modifications of Test No. 26
cold	hot		[gpm]	[gpm]	[gpm]	[mpg]	[$\frac{gal}{100 mi}$]	
×			0.27	2.15	0.12	23.7	4.22	
	×		0.08	0.29	0.10	24.4	4.10	1) without SA - and Spark Timing Switches with in the First Cycle
		×	0.05	0.11	0.08	32.4	3.09	2) without Switches
	×		0.08	0.21	0.11	25.0	4.00	1)
		×	0.05	0.27	0.09	32.6	3.07	2)
	×		0.07	0.50	0.13	24.6	4.07	1)
		×	0.05	0.36	0.08	32.4	3.09	2)
	×		0.07	0.28	0.10	24.8	4.03	1)
		×	0.04	0.08	0.06	32.5	3.08	2)
	×		0.06	0.04	0.13	24.7	4.05	1)
		×	0.05	0.006	0.08	32.3	3.10	2)
×			0.19	1.63	0.13	22.2	4.50	
×			0.18	1.55	0.14	23.3	4.29	
×			0.17	1.36	0.12	23.1	4.33	
×			0.18	1.70	0.14	23.4	4.27	

Tab.: 6.2.8

Modification Code	Cold Start Enrichment Serial	without Cold Start Injection at +20°C	without Spark Advance Diaphragm after Start for	Proportional EGR „On" after Cold Start	Control Pressure Hot Engine (3.55 Serial)	SA twisted from TWC to OXCAT after	Idle Spark Timing
17	"	"	90 sec	—	3.00 (richer as serial)	60 sec	3° ATDC
14	"	"	50 sec	immediately	3.55	70 sec	3° ATDC 60" after cold start without spark retard diaphragm
13	"	"	50 sec	—	3.55	50 sec	3° ATDC 70" after cold start without spark retard diaphragm
15	"	"	50 sec	—	3.90	—	3° ATDC 90" after cold start without spark retard diaphragm
20	"	"	80 sec	70 sec	2.50	80 sec	8° ATDC
16	"	"	80 sec	90 sec	3.40	80 sec	3° ATDC
18	"	"	90 sec	immediately	3.00	60 sec	3° ATDC
19	"	"	80 sec	—	3.55	—	3° ATDC

Tab.: 6. 2. 9

Test	Modification Code	Regulated Emissions						Unregulated Emission	
		HC		CO		NOx		SO_4	
		Test	AV.	Test	AV.	Test	AV.	Test	AV.
UDC	01	3.59		55.0		1.79		1.13	
UDC	01	3.54		55.6		1.74		1.01	
UDC	01	3.23	3.39	50.2	54.6	1.75	1.79	0.82	0.94
UDC	01	3.42		56.5		1.75		0.89	
UDC	01	3.15		55.7		1.93		0.83	
SET *	01	2.18		39.7		1.95		1.19	
SET *	01	2.31		38.7		2.11		1.10	
SET *	01	2.11	2.18	38.3	40.0	2.00	2.05	1.03	1.08
SET *	01	2.33		42.1		2.05		0.96	
SET *	01	1.99		41.4		2.14		1.13	
HDC	01	1.53		32.5		1.99		1.20	
HDC	01	1.62		32.5		1.94		1.16	
HDC	01	1.58	1.57	34.1	33.8	1.87	1.92	1.20	1.21
HDC	01	1.63		34.6		1.86		1.10	
HDC	01	1.49		35.1		1.95		1.40	

Test	Modification Code	Regulated Emissions						Unregulated Emission	
		HC		CO		NOx		SO_4	
		Test	AV.	Test	AV.	Test	AV.	Test	AV.
UDC	05	0.77		7.80		2.78		0.63	
UDC	05	0.76		13.5		2.34		0.88	
UDC	05	0.79	0.76	14.3	12.6	2.30	2.43	0.85	0.62
UDC	05	0.70		12.1		2.45		0.35	
UDC	05	0.80		15.6		2.29		0.41	
SET *	05	0.29		3.77		3.11		0.61	
SET *	05	0.37		9.22		2.83		0.96	
SET *	05	0.35	0.33	6.78	6.47	2.89	3.00	0.54	0.55
SET *	05	0.32		6.39		3.19		0.33	
SET *	05	0.32		6.21		2.97		0.31	
HDC	05	0.14		0.77		3.82		0.46	
HDC	05	0.17		2.84		3.24		0.59	
HDC	05	0.17	0.16	2.40	2.17	3.30	3.43	0.46	0.45
HDC	05	0.17		2.51		3.58		0.36	
HDC	05	0.17		2.31		3.23		0.40	

Test	Modification Code	Regulated Emissions						Unregulated Emission	
		HC		CO		NOx		SO_4	
		Test	AV.	Test	AV.	Test	AV.	Test	AV.
UDC	09	0.52		9.69		0.81		0.46	
UDC	09	0.59		10.1		0.81		0.59	
UDC	09	0.70	0.60	10.5	9.61	0.80	0.85	0.50	0.54
UDC	09	0.60		8.13		0.91		0.71	
UDC	09	0.60		9.68		0.92		0.43	
SET *	09	0.14		4.83		1.08		0.68	
SET *	09	0.17		4.30		1.09		0.56	
SET *	09	0.20	0.18	4.90	4.49	1.09	1.17	0.38	0.50
SET *	09	0.17		3.49		1.31		0.51	
SET *	09	0.23		4.95		1.30		0.39	
HDC	09	0.06		0.93		1.03		0.69	
HDC	09	0.04		1.60		1.08		0.60	
HDC	09	0.04	0.05	0.97	1.09	1.14	1.19	0.49	0.55
HDC	09	0.04		0.83		1.34		0.56	
HDC	09	0.06		1.10		1.37		0.43	

Test	Modification Code	Regulated Emissions						Unregulated Emission	
		HC		CO		NOx		SO_4	
		Test	AV.	Test	AV.	Test	AV.	Test	AV.
UDC	17	0.35		4.12		0.15		0.55	
UDC	17	0.45		4.07		0.19		0.84	
UDC	17	0.46	0.45	4.62	4.24	0.18	0.17	0.54	0.58
UDC	17	0.49		4.14		0.18		0.55	
UDC	17	0.49		4.25		0.14		0.40	
SET *	17	0.12		1.13		0.09		0.71	
SET *	17	0.14		0.99		0.09		1.45	
SET *	17	0.17	0.15	1.43	1.22	0.09	0.09	0.78	0.79
SET *	17	0.14		1.09		0.12		0.64	
SET *	17	0.17		1.45		0.06		0.39	
HDC	17	0.13		1.19		0.04		0.79	
HDC	17	0.13		0.93		0.04		1.50	
HDC	17	0.15	0.14	0.91	1.08	0.02	0.04	0.96	0.94
HDC	17	0.15		1.26		0.04		0.85	
HDC	17	0.13		1.12		0.04		0.61	

* Sulphate Emission Test

Tab.: 6. 2. 10

Test	Modification Code	Regulated Emissions HC Test	HC AV.	CO Test	CO AV	NOx Test	NOx AV	Unregulated Emission HCN Test	HCN AV.
UDC	01	3.59		55.0		1.79		10.5	
	01	3.54		55.6		1.74		9.1	
	01	3.23	3.39	50.2	54.6	1.75	1.79	9.6	10.1
	01	3.42		56.5		1.75		8.6	
	01	3.15		55.7		1.93		12.9	
SET*	01	2.18		39.7		1.95		7.3	
	01	2.31		38.7		2.11		10.2	
	01	2.11	2.18	38.3	40.0	2.00	2.05	5.8	8.9
	01	2.33		42.1		2.05		12.4	
	01	1.99		41.4		2.14		8.8	
HDC	01	1.53		32.5		1.99		3.0	
	01	1.62		32.5		1.94		10.2	
	01	1.58	1.57	34.1	33.8	1.87	1.92	9.9	6.5
	01	1.63		34.6		1.86		5.1	
	01	1.49		35.1		1.95		4.2	

Test	Modification Code	Regulated Emissions HC Test	HC AV.	CO Test	CO AV	NOx Test	NOx AV	Unregulated Emission HCN Test	HCN AV.
UDC	05	0.77		7.80		2.78		5.8	
	05	0.76		13.5		2.34		16.0	
	05	0.79	0.76	14.3	12.6	2.30	2.43	12.4	12.5
	05	0.70		12.1		2.45		13.5	
	05	0.80		15.6		2.29		14.7	
SET*	05	0.29		3.77		3.11		2.4	
	05	0.37		9.22		2.83		9.0	
	05	0.35	0.33	6.78	6.47	2.89	3.00	6.6	6.2
	05	0.32		6.39		3.19		5.7	
	05	0.32		6.21		2.97		7.2	
HDC	05	0.14		0.77		3.82		2.3	
	05	0.17		2.84		3.24		5.3	
	05	0.17	0.16	2.40	2.17	3.30	3.43	5.5	4.4
	05	0.17		2.51		3.58		2.9	
	05	0.17		2.31		3.23		5.9	

Test	Modification Code	Regulated Emissions HC Test	HC AV.	CO Test	CO AV.	NOx Test	NOx AV	Unregulated Emission HCN Test	HCN AV.
UDC	09	0.52		9.69		0.81		3.4	
	09	0.59		10.1		0.81		3.3	
	09	0.70	0.60	10.5	9.61	0.80	0.85	6.9	5.4
	09	0.60		8.13		0.91		7.6	
	09	0.60		9.68		0.92		5.8	
SET*	09	0.14		4.83		1.08		2.0	
	09	0.17		4.30		1.09		2.6	
	09	0.20	0.18	4.90	4.49	1.09	1.17	3.0	2.7
	09	0.17		3.49		1.31		3.1	
	09	0.23		4.95		1.30		3.0	
HDC	09	0.06		0.93		1.03		0.8	
	09	0.04		1.60		1.08		2.0	
	09	0.04	0.05	0.97	1.09	1.14	1.19	2.3	2.2
	09	0.04		0.83		1.34		2.3	
	09	0.06		1.10		1.37		3.7	

Test	Modification Code	Regulated Emissions HC Test	HC AV.	CO Test	CO AV.	NOx Test	NOx AV	Unregulated Emission HCN Test	HCN AV.
UDC	17	0.35		4.12		0.15		4.2	
	17	0.45		4.07		0.19		1.8	
	17	0.46	0.45	4.62	4.24	0.18	0.17	4.0	3.3
	17	0.49		4.14		0.18		4.3	
	17	0.49		4.25		0.14		2.4	
SET*	17	0.12		1.13		0.09		0.63	
	17	0.14		0.99		0.09		0.34	
	17	0.17	0.15	1.43	1.22	0.09	0.09	1.13	6.6
	17	0.14		1.09		0.12		0.76	
	17	0.17		1.45		0.06		0.42	
HDC	17	0.13		1.19		0.04		0.63	
	17	0.13		0.93		0.04		0.83	
	17	0.15	0.14	0.91	1.08	0.02	0.04	0.84	0.86
	17	0.15		1.26		0.04		0.79	
	17	0.13		1.12		0.04		1.01	

* Sulphate Emission Test

Tab.: 6. 2. 11

	Modification Code	01	03	04	05	07	08	09	11	12	13	15	16
	License Plate	WOB-VA 67	WOB-VX 36	WOB-VR 93	WOB-VD 93	WOB-VX 36	WOB-VR 93	WOB-VD 93	WOB-VX 36	WOB-VR 93	WOB-VD 50	WOB-VS 35	BS-JV 667
Exterior Noise	Idle	70	71	71	76	72	71.5	77.5	71.5	72	69	74	70.5
	Cruise 30 mph	62.5	62	60.5	62.5	62	61.5	62	61.5	61.5	61	61	60.5
	Acceleration from 30 mph	73.5	74	72.5	72	73	73	72	72	73.5	72.5	71	71.5
Interior Noise	Idle	56	56	49	59	56	51	57	55.5	50.5	54	54	49
	Cruise 30 mph	63	65.5	61	64	65.5	61.1	64.5	65	59.5	63.5	63	58
	Acceleration from 30 mph	74.5	81.5	73.5	75.5	78.5	74.3	77	78.5	75	78	73	73

Tab.: 6. 2.12

Pick Up Position	Piston Direction [$\frac{m}{sec^2}$]				Transversal [$\frac{m}{sec^2}$]				Crankshaft Direction [$\frac{m}{sec^2}$]	
	2500 RPM		5000 RPM		2500 RPM		5000 RPM		2500 RPM	5000 RPM
	1.3 l	1.6 l	1.3 l	1.6 l	1.3 l	1.6 l	1.3 l	1.6 l	1.6 l	1.6 l
1	18.0	32.0	90.0	130.0	3.6	4.6	42.0	8.0	2.5	5.8
2	15.0	64.0	80.0	80.0	2.9	8.0	32.0	90.0	7.0	220.0
3	15.0	50.0	75.0	85.0	2.2	24.0	28.0	20.0	9.0	13.0
4	13.0		70.0		2.2		18.0			
5	11.0		65.0		1.9		15.0			
6	11.0	19.0	59.0	110.0	3.0	2.9	21.5	5.0	4.0	32.0
7	12.0	10.0	70.0	70.0		2.0	28.0		2.0	7.0
8					10.0		58.0			
9	9.0	5.0	52.0	31.0	1.1	2.6	17.0	29.0	3.0	12.0

Tab.: 6. 2. 13

6.3 CONSUMER ACCEPTABILITY

6.3.1 Methodology

The consumer acceptability of the engine/vehicle systems investigated under this Contract is an agglomeration of the following consumer attributes: Startability, driveability, acceleration performance, gradeability, and cost.

Startability and driveability were investigated simultaneously, the procedure being as follows: To begin with, measuring instruments monitoring oil temperature, engine speed, and engine torque are placed on the engine and connected to a recording instrument, the torque being derived from measuring the force exerted on one of the engine mounts.

Then, the temperature of the entire vehicle is brought to the point at which startability and driveability are to be investigated. When the temperature of the vehicle has stabilized it is transferred to a chassis dynamometer, where an expert driver puts it through the following starting and driving procedure:

1) Start according to operating instructions.

2) Idle for 30 seconds, then tip in so that the automatic choke returns to the normal operating position.

3) Engage first gear and start; beginning with 1,500 rpm open the throttle valve within 2 seconds and accelerate to 4,000 rpm; declutch.

   Engage second gear and clutch; accelerate again to 4,000 rpm by opening the throttle within 2 seconds; declutch.

   Engage third gear; accelerate to 3,500 rpm by opening the throttle within 2 seconds.

   Stop the vehicle after acceleration in third gear has ended; declutch at about 2,000 rpm. This terminates the first driving cycle.
   In case of automatic transmission in position "D" the drivetrain must be made to switch gears by briefly releasing the accelerator at 4,000 rpm.

4) This followed by the bucking test:
   Start in first gear; shift to second; drive at level speed of 1,500 rpm; then accelerate from 1,500 to 3,000 rpm within 25 seconds.

5) Repeat the driving cycle described under 3) until the engine oil temperature has reached approximately 60 °C.

6) Acceleration from coasting: Proceed slowly from coasting to full throttle within 5 seconds.
Initial coasting speeds:

Manual drivetrain:	3rd gear at 30, 40, 50 mph, 2nd gear at 20, 30, 40 mph.
Automatic:	3rd speed at 50, 60, 70 mph, 2nd speed at 25, 35, 45 mph.

7) After this part of the program, begin the sudden-acceleration-from-level-speed test, which means that after driving at constant speed the accelerator is dipped to full throttle within one second.

Initial speeds for this test:

Manual drivetrain:	2nd gear at 30, 40, 50 mph, 3rd gear at 40, 50, 60, 70 mph.
Automatic:	2nd speed at 30, 40, 50 mph, 3rd speed at 50, 60, 70 mph.

8) By now, the engine oil temperature has reached 80 - 90 °C. At this stage, another acceleration cycle as described under 3) is run.

Directly after this driveability cycle, the driver of the vehicle evaluates the printouts of the recording instruments (for a sample printout, see 6th Monthly Progress Report, page 52 - 87, Appendix) according to the VW evaluation system (Table 6.3.1). The result is an evaluation record as presented in Table 6.3.2. Together with their graphs, all evaluation records are collected in the Appendix (see Fig. 6.3.1).

The individual phases of the driveability cycle, the evaluation of which is to be found in column 3 of the evaluation record, are gathered together in engine temperature ranges, i.e. start-up phase, cold idle, acceleration under start-up conditions, first warm-up, second warm-up, and hot-engine phase. The difference between the first and second warm-up phases is that during the first warm-up phase the engine speed at idle is still at a high cold-start level.

The average results obtained from these engine temperature ranges are then listed in column 4 (Table 6.3.2) and weighted by the constant factors given in column 6 of Table 6.3.2. From this, the contribution of each engine temperature range to the overall result is computed (last column of Table 6.3.2). The sum total of all contributing factors forms the overall driveability rating of the vehicle in question under a certain set of environmental conditions.

It is a matter of course that the aggregate values resulting from an evaluation procedure of this kind can only be regarded with a bit of scepticism. For instance, a vehicle which is extremely difficult to start either hot or cold may very well display a good aggregate value, if it has good idling, and good acceleration after cold start and good driveability after the engine has come up to temperature. Yet a vehicle with these properties could be inacceptable, simply because it is too difficult to start.

It is, therefore, appropriate to evaluate the starting phase alone under the heading of startability. Moreover, it seems imperative to study each engine temperature range and compare it to the others. Only after this it is possible to evaluate the aggregate results without running the risk of going wrong.

The startability rating of each engine modification is given in Chapter 6.3.2 as determined by this method. All other engine temperature ranges will be evaluated in Chapter 6.3.4.

The ability of all vehicles to accelerate and pass both at low and high speeds was investigated according to the procedure given in the Federal Register, Vol 34, No. 99, 5-23-1969, using engine maps and vehicle data.

The passing ability of a vehicle is expressed by the time in seconds and the distance in feet which it would hypothetically require to pass another vehicle 55 feet long, traveling either at 20 or at 50 mph, under the following conditions:

1) The vehicle weight is at GVWR, except that the fuel tank may be filled to any level between 90 and 100 percent of its capacity.

2) The following accessories and equipment are operating at maximum power: air conditioner, or heater if vehicle is not equipped with an air conditioner, windshield wipers, and headlamps on high beam.

3) Ambient temperature is between 59 °F and 85 °F, ambient dry barometric pressure is between 28,50 in. hg. and 20,50 in. hg. and relative humidity is between 30 percent and 60 percent.

4) The road has a grade of zero percent and the road surface has a skid number of 75.

5) Wind velocity is zero.

The data represent the performance capability of the vehicle in performing the two hypothetical maneuvers described below:

The vehicle about to pass follows another vehicle (pace vehicle) which is 55 feet long, with the leading edge of the passing vehicle 40 feet behind the trailing edge of the pace vehicle and both vehicles traveling at 20 mph. The pace vehicle travels at constant speed throughout. The passing vehicle is in a different lane from the pace vehicle. The passing maneuver begins when the passing vehicle accelerates at its maximum rate without use of clutch or brake up to a speed limit of 35 mph or to its maximum speed (if less than 35 mph). It maintains that speed, or maximum acceleration if unable to reach either the limiting or maximum speed, until the maneuver ends when its trailing edge is 40 feet ahead of the leading edge of the pace vehicle.

The other maneuver is the same as before with an initial speed of 50 mph (instead of 20 mph), a speed limit of 80 mph (instead of 35 mph), and separation of 100 feet (instead of 40 feet) before and after the maneuver (see Fig. 6.3.2).

Vehicle gradeability was computed as well. These computations are based on the full-load curve derived from engine map measurements and on data concerning vehicle mass, the transmission ratios of the various gears, and the efficiency of the drivetrain used.

The results of these computations are listed in Chapter 6.3.5 and presented in the Appendix in the manner of which Fig. 6.3.3 is a sample.

6.3.2 Startability

Table 6.3.3 lists the startability ratings of all engine system modifications investigated under this Contract. The figures listed in this Table are presented as graphs in Figs. 6.3.4 and 6.3.5. Fig. 6.3.4 shows the startability of the vehicles at temperatures of -10 °C, +10 °C, and +30 °C at sea level, whereas Fig. 6.3.5 describes their startability at an altitude of 1600 m and a temperature of +10 °C. The latter shows for purposes of comparison the startability found at the same temperature at sea level, too.

The first thing which meets the eye in Fig. 6.3.4 is that the startability of Modification Codes 05 and 09, 06 and 10, 07 and 11, and 08 and 12 is the same. This is not fortuitous; it is because in these Modification pairs, vehicles and engines are the same, and because from one Modification to the next, the emission control concepts change only in a manner which does not affect startability. For this reason, we only measured the startability of Modification Codes 09 through 12 and attributed the appropriate figures to Modifications 05 through 08.

It is remarkable to note the discrepancies in startability between Modification Codes 09 and 10 and between 17 and 18. These discrepancies are due to scatter, for in both cases the engines are completely identical, the only difference being that Codes 05 and 17 involve an inertia weight of 2,250 lbs, whereas 06 and 18 have 3,000 lbs. This, however, cannot affect startability in any way.

Therefore, as far as startability tests of this kind are concerned, it seems probable that we shall have to live with scatter bandwidths extending across two full grades, and it would seem natural to have not one but many starting tests with each Modification Code to consolidate our findings. But even then, the discrepancy in the startability at +30 °C between Modification Codes 09 and 10 is unusually large. We suppose this to be due to bias of some kind which probably occurred when testing Code 10, because the startability at +30 °C is clearly inferior generally to that at +10 °C.

None of the Modifications received a startability rating of less than 6, which is a pass according to Table 6.3.1. Still, those Modifications which received ratings below 8 may well cause difficulties, because there is always the possibility of downward scatter.

Problems of this kind, however, will occur only in carburetor and not in fuel injection engines, where the lowest startability rating ever awarded was 8. All other ratings are better than this, so that we may state generally that the startability of these Modifications is superior to that of the Uncontrolled Modifications as well as of those meeting the '76 Standards.

In Fig. 6.3.4, there are again discrepancies in the startability at high altitudes of Modifications 01 and 02 as well as 13 and 14. These discrepancies are due to normal scatter.

However, the worse high-altitude startability of Modifications 17 through 19 is not due to scatter; it is due to the fact that while these measurements were taken we had deactivated the altitude sensor so as to find a proper tuning for meeting the Research Standard and stabilizing the CO emissions. This situation continued unchanged throughout the driveability tests, which is why the startability of the fuel injection engines at high altitude is clearly inferior.

6.3.3 Driveability

The form sheets on which the driveability rating of each Modification Code was evaluated are included in the Appendix. Table 6.3.4 shows all figures listed in column 4 of the evaluation records, i.e. the average performance ratings referring to the various engine temperature ranges. The only rating not given here is that of the startability because it has already been presented in Table 6.3.3.

From the figures listed in Table 6.3.4, Figs. 6.3.6 through 6.3.11 have been drawn. Figs. 6.3.6 through 6.3.10 are comparisons of driveability within the various engine temperature ranges, whereas Fig. 6.3.11 compares the driveability aggregates. Unlike our startability graphs, where the low and high altitude ratings were entered in separate figures, the graphs here show the two together, with the fat lines indicating high-altitude driveability.

These diagrams are easily more informative than the startability graphs. True, all the figures used here were obtained from isolated tests which were not repeated. On the other hand, all figures as a rule are averages derived from a number of evaluation phases, which makes them statistically more secure than the figures pertaining to startability. This, of course, applies especially to the aggregate values given. It is even possible that they may run to the other extreme in that they may cover up isolated negative phenomena by averaging them out.

The most important result to be obtained from a quick survey of all 6 figures is that, on an average, the driveability of the fuel injection engines is best all around. It follows that engine concepts of this kind do not merely furnish low emissions and good fuel consumption but good driveability as well. Of course, this is not ascribable to low emission standards; but in order to meet these low emission standards, an extremely expensive and sophisticated carburation system (K-Jetronic) is required, and this produces good driveability. This applies at all ambient temperatures and altitudes, although when tested at high altitude the K-Jetronic engines had a few problems due to the altitude sensors being deactivated.

The gravest driveability problems occur in all vehicles and especially in those equipped with carburetors under starting conditions, during the first acceleration, the bucking test, and the first warm-up phase. Fig. 6.3.7, where driveability under starting conditions is compared, shows that driveability has deteriorated under the '76 California Standard. This effect continues to be noticeable throughout the first warm-up phase, after which it balances out.

The vehicles which received the most inferior ratings of all are Modification Codes 08, 11, and 12; 08 and 12 being a 3,000 lbs vehicle equipped with a tuned-up 1.6 l engine, whereas 11 is the Rabbit with a 1.3 l engine. It is surely due to some accident that those engines which are relatively small compared to their vehicle mass have received such bad ratings. After all, there is no reason why the driveability of smaller engines should necessarily be bad. The explanation is probably that in all three cases the engines are not standard engines, i.e. they were originally destined for the European market and had been made to meet the '76 US Federal and California Standards by being retuned and by having a few auxiliary devices, such as self-aspirating secondary air systems and catalysts, added to them. It is quite natural that this should entail some loss of driveablilty.

It is important, however, that the cumulative driveability of these three vehicles is hardly affected by the bad driveablity ratings they received at start-up and during the first warm-up phase. True, their cumulative driveability is bad as well, but it is not as outstandingly bad.

Looking at the cumulative driveability ratings one might say that Uncontrolled engines as well as engines meeting the '76 Federal Standard work best at an ambient temperature of +30 °C, whereas automobiles destined for California prefer +10 °C. It follows that -10 °C is the temperature at which the driveability of all Modifications is at its worst.

All this is due to exhaust emission regulations, for all vehicles, even those equipped with Uncontrolled engines, are tuned to minimum CO emissions, and this is bound to cause a certain loss of driveability, especially if the engine is cold.

However, we are sure that the bad driveability of Modification Code 11 at -10 °C even with the engine warmed up - bad enough to result in the only driveability rating below 6, see Fig. 6.3.10 - is merely accidental. We suppose this to be due to faulty measuring.

6.3.4 Acceleration Performance

Based on engine maps and vehicle data, acceleration performance was computed according to the passing ability calculation procedure described in the Federal Register, Vol. 34, No. 99, 5-23-1969. The results of these computations are given in Table 6.3.5.

Looking at these results, it is important first of all to keep in mind that all these are tests of single cars, production dispersion not being taken into account. And because these were run on random vehicles, their acceleration performance may be either superior or inferior to that of the average production vehicle.

As all the engine maps from which the data for the acceleration performance computations were derived were generated from engines not equipped with catalysts, the acceleration performance of a number of Modification Codes is identical. Thus, the acceleration performance of Modification Codes 05 and 01 has to be identical because there is no way in which a spark-retard diaphragm and a self-aspirating secondary air system might influence acceleration (see Table 4.1.2). The same applies to Modification Code 07, the acceleration performance of which must be identical with that of Modification 03. Furthermore, the acceleration performances of Modification Codes 06 and 10 and those of 08 and 12 must be identical as well. Consequently, we do have acceleration performance figures for all Modifications, although Modifications 05 through 08 have not been tested and have no ratings ascribed to them in Table 6.3.5.

Figs. 6.3.12 through 6.3.15 are presentations of the passing ability of comparable engines as given in Table 6.3.5. The shorter the bars are in these diagrams, the better is the passing ability of the vehicle in question, for the length of the bars indicates the distance or the time required to complete the passing maneuver. Save for Fig. 6.3.14, where the Rabbits equipped with the 1.3. l engines are compared, it may be said that fuel injection engines, i.e. modifications with low emission standards, display a passing ability just as good or even better than Uncontrolled vehicle. It is natural that this should come out most distinctly in Fig. 6.3.15, where the fuel injection side is represented by 5-cylinder 2.2. l engines competing against 4-cylinder 1.6 l engines (Modifications 04, 08, and 12). The superiority of powerful engines is much more marked when passing at high speed than when passing at low speed.

Why it is that each of these diagrams shows that all vehicles meeting the '76 Standard are clearly inferior as far as at least their high-speed passing ability is concerned cannot be explained by any patent data, such as peak horsepower, maximum torque, inertia weight, or transmission ratios. It may be due to the fact that vehicles destined for the U.S. market incorporate some pieces of equipment which Uncontrolled vehicles do not have.

Fig. 6.3.16 again shows all passing times versus the peak-horsepower-to-inertia-weight ratios. There are three reasons why there can be no linear relationship here:

1) The air drag differs from one vehicle to another (see Table 5.2.2);

2) The transmission ratios differ from one vehicle to another (see Table 5.2.2); and, finally,

3) The full-load-over-engine-speed curves also differ from one vehicle to another (see Appendix).

Still, we did find that there is quite a close correlation between the passing time at high speed and the peak-horsepower-to-inertia-weight ratio. This is not surprising, for the passing time at high speed is measured with the power output of the engine coming much closer to the peak horsepower than it does during the low-speed passing maneuver.

The same figure also shows quite distinctly that comparable modifications are clustered together, and that the passing times at low and at high speed differ less and less, the more the peak-horsepower-to-inertia-weight ratio increases.

This last circumstance necessarily results from the fact that the procedure of measuring the low-speed and high-speed passing time is always the same, independently of the peak-horsepower-to-inertia-weight ratio. However, the higher the HP/IW ratio is, the greater will be the superiority over vehicles with a lower HP/IW ratio when running through a long-drawn-out acceleration process (high-speed passing) compared to a brief acceleration process (low-speed passing).

6.3.5 Gradeability

Table 6.3.5 lists the results of our gradeability computations together with the acceleration performance figures. Gradeability was computed according to the procedure described in Chapter 6.3.1.

All performance diagrams generated are included in the Appendix. Table 6.3.5 lists the maximum gradeability theoretically attainable in each gear, no matter whether it is actually feasible or not.

Quite a number of the first-gear gradeability ratings given in this Table are not feasible in actual practice because under normal conditions the wheels would slip. With the vehicle at half payload, the friction coefficient is normally assumed to be 0.8. In that case, the vehicle will climb slopes of up to approximately 38 % inclination before the wheels begin to slip. Therefore, all first-gear ratings exceeding 38 % merely indicate what is possible theoretically, but these ratings cannot be implemented due to the physical laws governing the interaction between tire and road.

Here again, we have to reiterate what we said when dealing with acceleration performance: All gradeability figures are ratings pertaining to isolated cars; production dispersion is not taken into account.

In Figs. 6.3.17 through 6.3.20, all gradeability figures from Table 6.3.5 have been broken down into comparable engine/vehicle systems. The trends visible here deviate clearly from those indicated by the passing ability diagrams, because gradeability is governed by practically nothing but the maximum torque of the engine. We demonstrated this in Fig. 6.3.21, which is a presentation of the gradeability of comparable engine families over maximum torque in all gears. The relationship is not always entirely linear, which is due to the fact that because of the constant transmission ratios, maximum inclinability and maximum torque do not necessarily coincide in every case. Still, gradeability in all gears is so decisively influenced by maximum torque that it would be just as well to judge gradeability from Table 5.2.2, where the maximum torque figures are listed.

Here again, we can see that K-Jetronic engines perform quite well, and that there is no question about their being comparable to engines meeting easier exhaust emission standards. It goes without saying that from all K-Jetronic engines the gradeability of the 5-cylinder engines is most outstanding.

6.3.6 Cost of Engine Systems

VW's accounting system differs from the break-down system of U.S. companies and does not permit allocating cost to a specific project. In our cost accounting system, there is no way for us to show up the cost of individual system components.

This is mainly due to one basic difficulty, that VW does not develop any components, but only concepts. Although there are some rare exceptions where development projects referring to individual devices are initiated, what we develop as a general rule are engine concepts related to concrete applications in the field, which means that there are always goals defining power output, torque, fuel economy, and emission targets, and the engines and all their auxiliaries are developed to meet those targets. After the completion of such a development process we know the development, investment, and manufacturing cost of the engine concept, but we do not know what device accounted for what percentage of the cost total.

It is for this reason that we are not in a position to state the cost of each Modification analysed under this Contract, for out of the 20 Modifications analysed there are only 5 standard products of which we know the cost.

Another difficulty is presented by the unstable exchange rate between the Dollar and the Deutsche Mark, for none of the standard products is being marketed in the U.S. today.

So as to provide, in spite of this difficulty, some data concerning the cost of the various engine concepts we took a survey of the market prices which our U.S. service organization is asking for the assemblies used by us or, alternatively, for similar assemblies. From the data thus collected we computed the total service prices of all Modifications as well as their prices relative to each other.

As a baseline for this comparison, we chose the Uncontrolled-1.6 l Rabbit engine (Modification Code 01) and put it in relation to the prices of all other Modifications. We thus arrived at the figures listed in column 1 of Table 6.3.6. Let us take a look at two of these figures: The 1.3 l Uncontrolled engine (Modification Code 03) is cheaper by about 10 % than the 1.6 l Uncontrolled engine, and the 2.2 l 5-cylinder engine, which is capable of meeting the '81 emission goals, is about twice as expensive as the 1.6 l Uncontrolled engine.

Moreover, Table 4.1.2 also shows that there is no difference between Modifications belonging to the same engine families which meet the emission tasks of 0.41/3.4/1.0 and 0.41/3.4/0.4, the sole exception being the 1.3 l engine (Modification 19), which had to be fitted with a deceleration control valve. However, the influence of this installation on cost is slight. Therefore, we may say that there is no difference in the cost of Modification Codes 13 and 17, 14 and 18, 15 and 19, and 16 and 20. This is why we grouped them all together in Table 6.3.6.

For the purpose of assessing the price increase caused by more stringent emission standards, only the following Modifications may be compared:

01 and 05, 09, 13/17 (1.6 l Rabbit, 2,250 lbs)
02 and 06, 10, 14/18 (1.6 l Rabbit, 3,000 lbs)
03 and 07, 11, 15/19 (1.3 l Rabbit, 2,250 lbs)
04 and 08, 12, 16/20 (1.6/2.2 l Audi 100/5000, 3,000 lbs)
so that Modifications 01, 02, 03, and 04 furnish the baseline prices.

These figures have been put in relation to each other in Table 6.3.6.

This operation shows that the introduction of emission control has led to a price increase of around 20 % in all concepts, with the differences between the '76 Federal and the '76 California concept being mainly due to tuning, which is the cause of nearly the same system cost but of the relatively high fuel consumption of the California concept.

Expressed in relative figures, the cost of the small engines increases somewhat more sharply, for the basic price of these engines is lower, yet the additional expenditure required to make them meet the emission standards is about the same as that of larger engines.

However, as far as meeting the future emission standards is concerned the expenditure on this particular engine is not quite as high in absolute as well as in relative figures, because the modifications required are relatively small in scope. The cost increase here amounts to a mere 40 %.

The cost of the 1.6 l Rabbit engine, on the other hand, will increase by about 60 %, whereas the cost of the Rabbit engine intended for the larger inertia weight will even grow by 70 %, mainly because a complicated EGR system and a secondary air pump will have to be installed.

As far as the higher performance engine is concerned, the transition from today's to tomorrow's emission concepts is accompanied by a changeover from the 4-cylinder to the 5-cylinder engine. Yet the cost increase expressed in relative figures is about as high as that of the Rabbit engine, because the standard to which it is related is that of a relatively expensive engine.

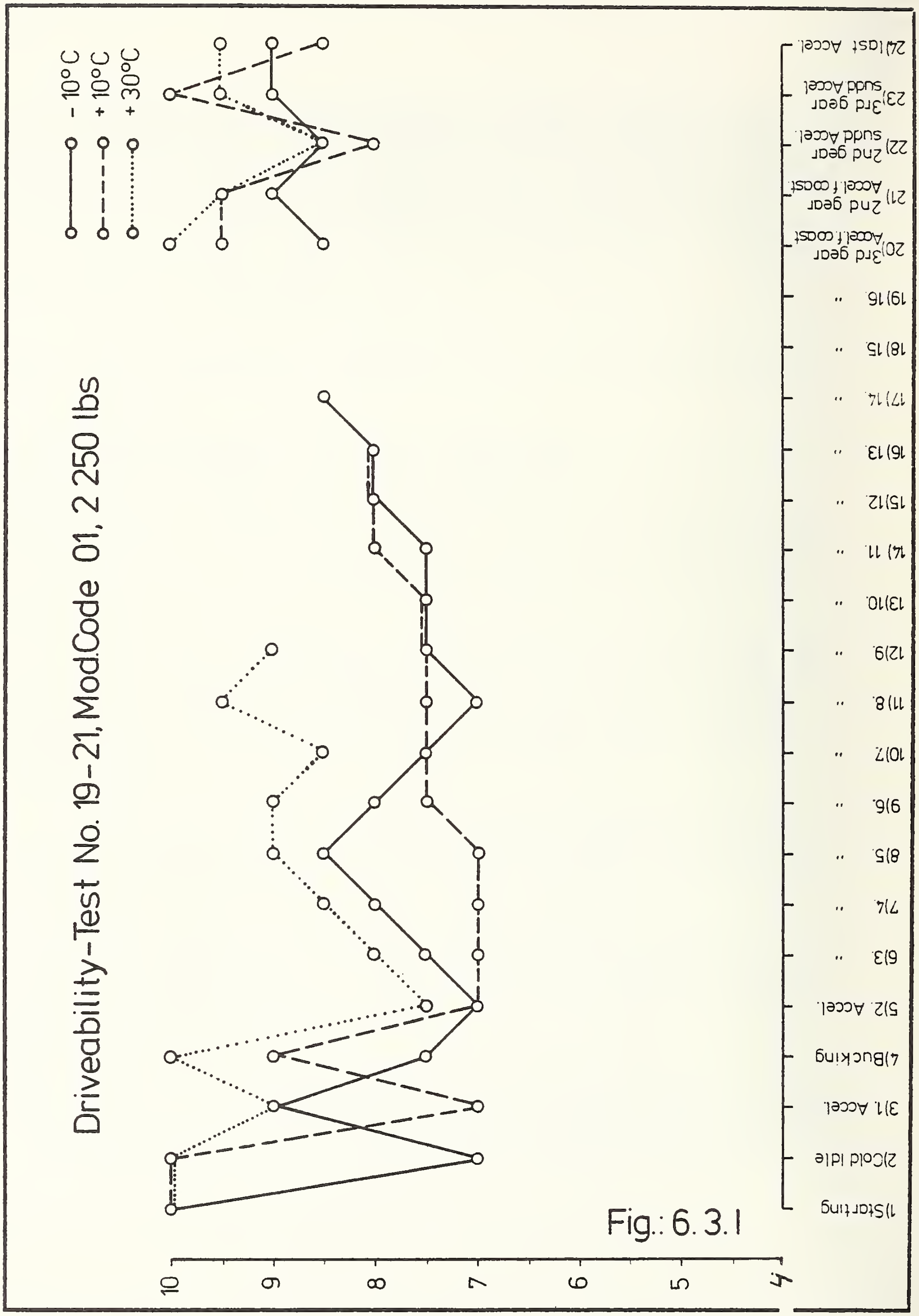

Driveability-Test No. 19-21, Mod.Code 01, 2 250 lbs
- 10°C
+ 10°C
+ 30°C
10
9
8
7
6
5
4
1)Starting
2)Cold idle
3)1. Accel.
4)Bucking
5)2. Accel.
6)3. "
7)4. "
8)5. "
9)6. "
10)7. "
11)8. "
12)9. "
13)10. "
14) 11. "
15)12. "
16)13. "
17)14. "
18)15. "
19)16. "
20)3rd gear Accel. f. coast
21)2nd gear Accel. f. coast
22)2nd gear sudd Accel.
23)3rd gear sudd Accel.
24)last Accel.

Fig.: 6.3.1

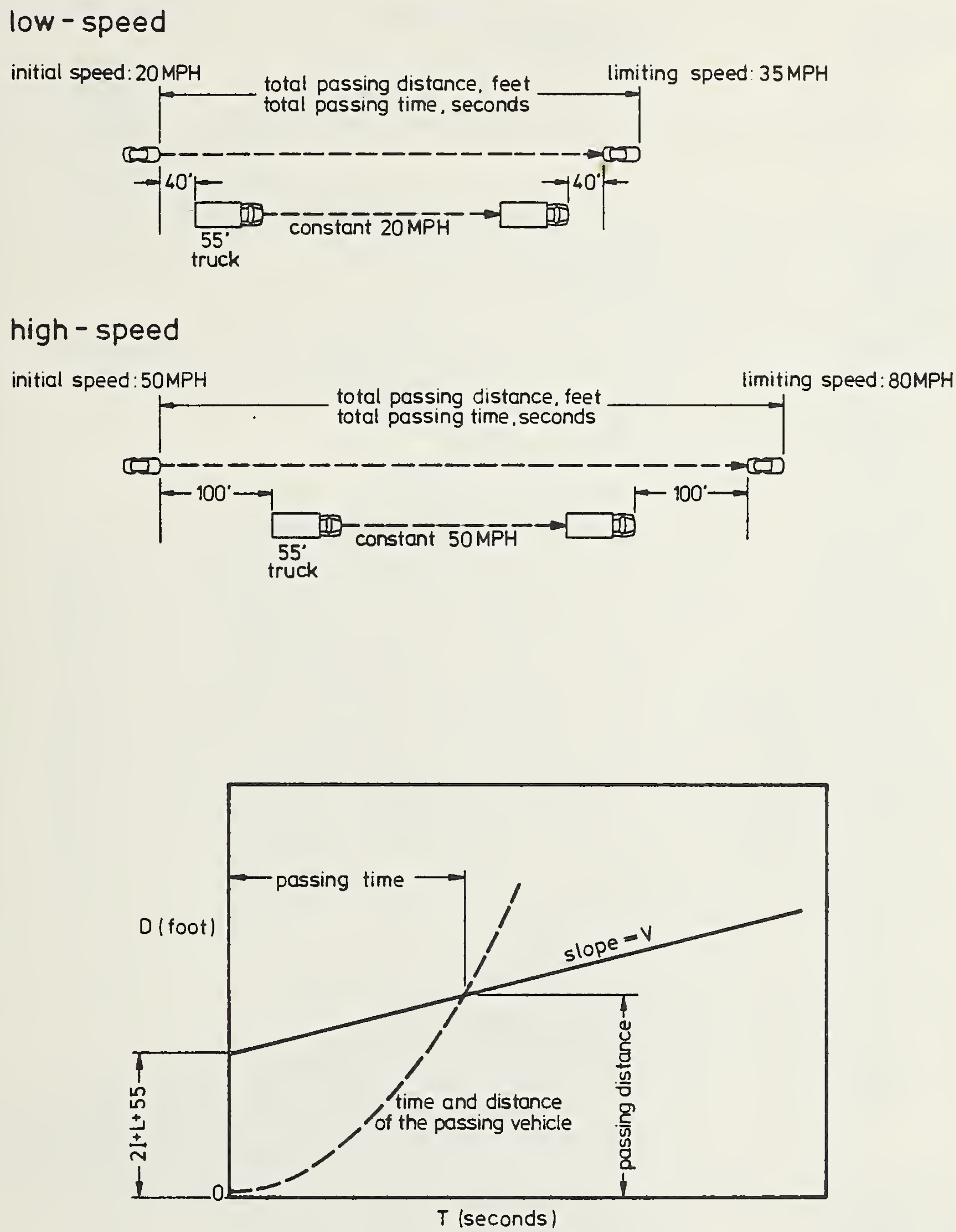

low - speed
initial speed: 20 MPH
total passing distance, feet
total passing time, seconds
limiting speed: 35 MPH
40'
40'
55' truck
constant 20 MPH
high - speed
initial speed: 50 MPH
total passing distance, feet
total passing time, seconds
limiting speed: 80 MPH
100'
100'
55' truck
constant 50 MPH
passing time
D (foot)
slope = V
2l+L+55
0
time and distance of the passing vehicle
passing distance
T (seconds)

Fig.: 6.3.2

PKW-ENTW. BERECHNUNG	FAHRLEISTUNGSDIAGRAMM TYP 17 RABBIT MOTOR 01	B .171.000-161 BLATT: 147
		DATUM: 25.01.78 SIGNUM: Mull
		ANLAGE ZUM BERECHNUNGSBERICHT T
		EA 718

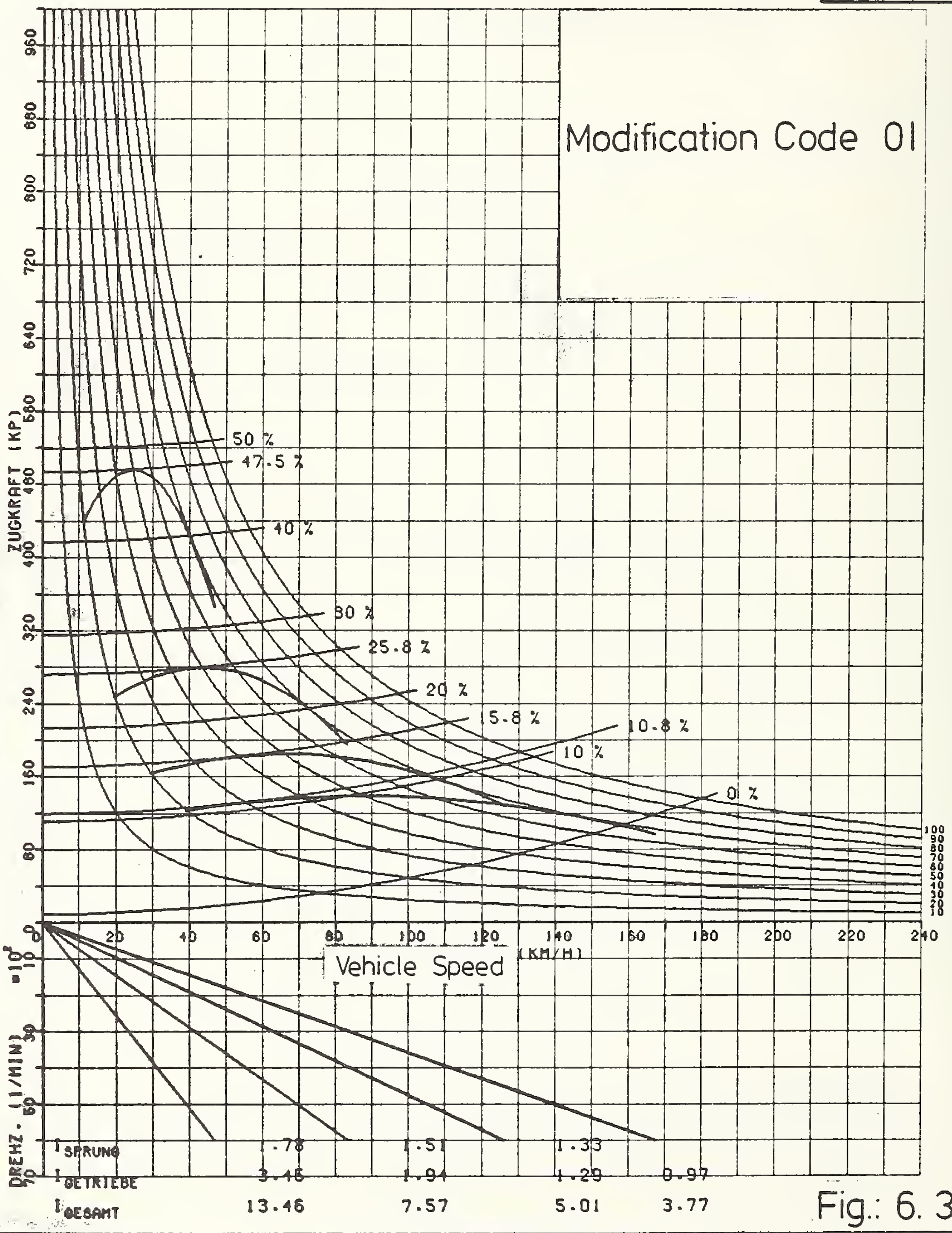


Fig.: 6. 3. 3

EPEB12R 25.01.78 09.08.48 0001

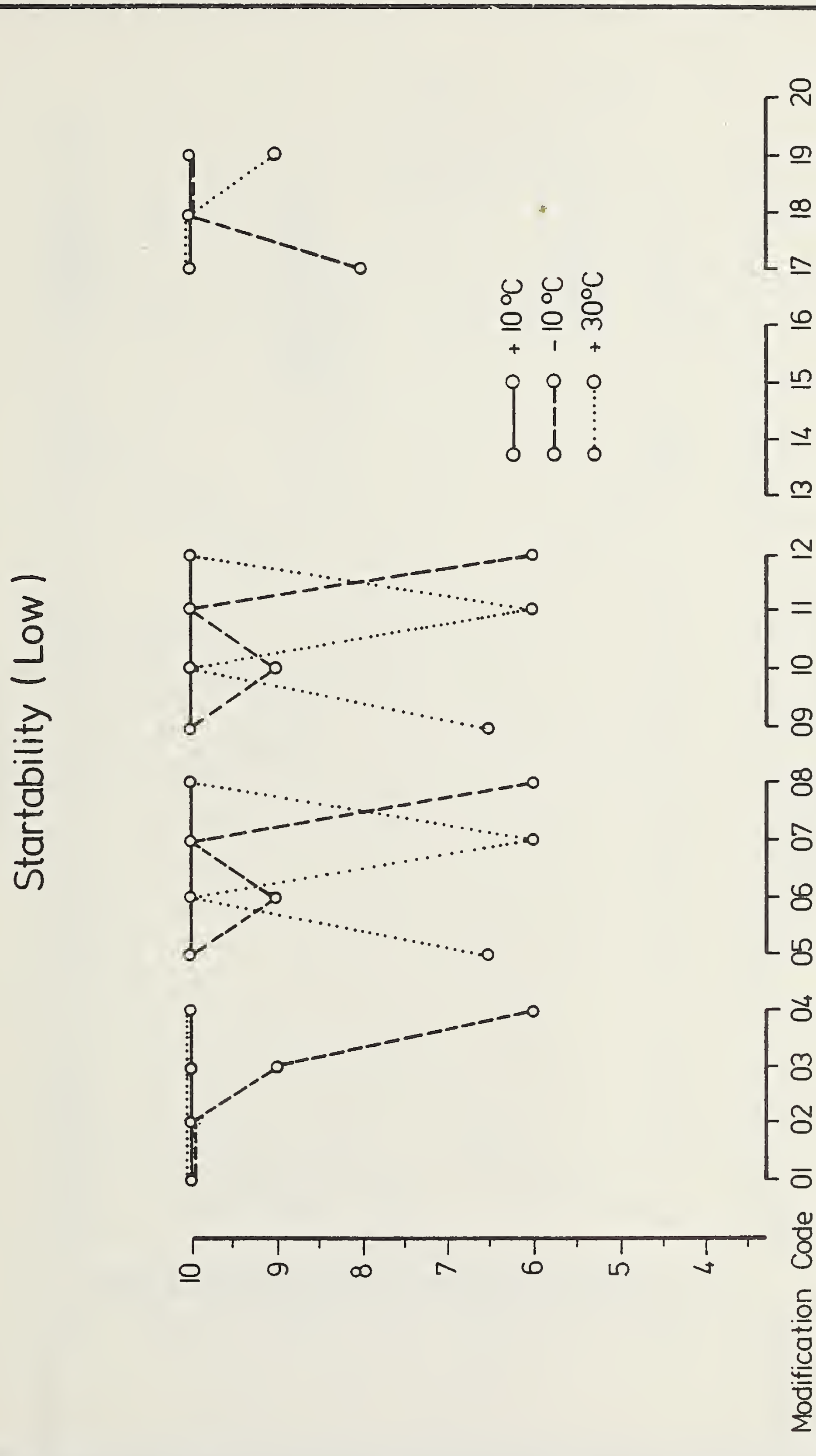

Startability (Low)
10
9
8
7
6
5
4
Modification Code
01 02 03 04 05 06 07 08 09 10 11 12 13 14 15 16 17 18 19 20
+ 10°C
- 10°C
+ 30°C

Fig.: 6. 3. 4

Startability (+10°C)

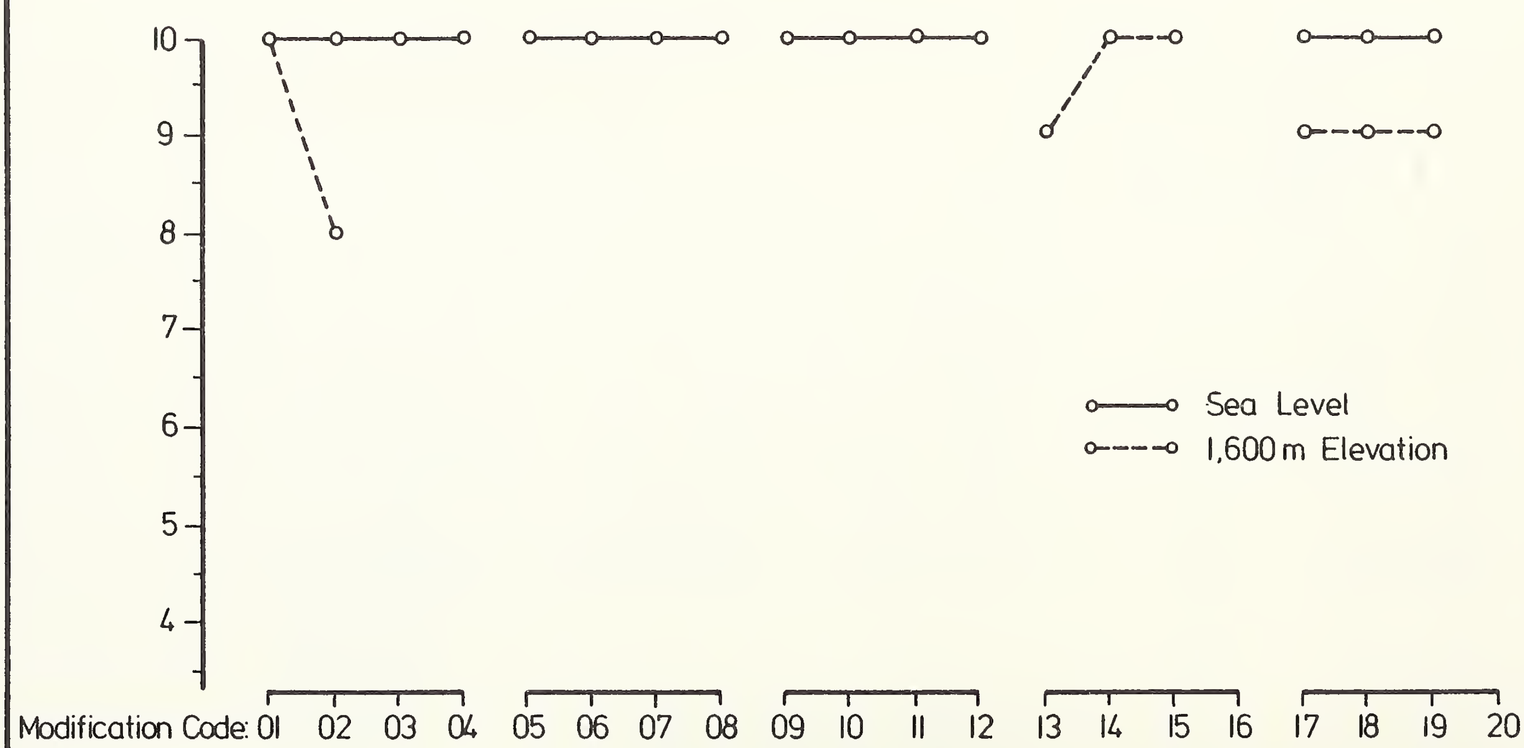


Fig.: 6. 3. 5

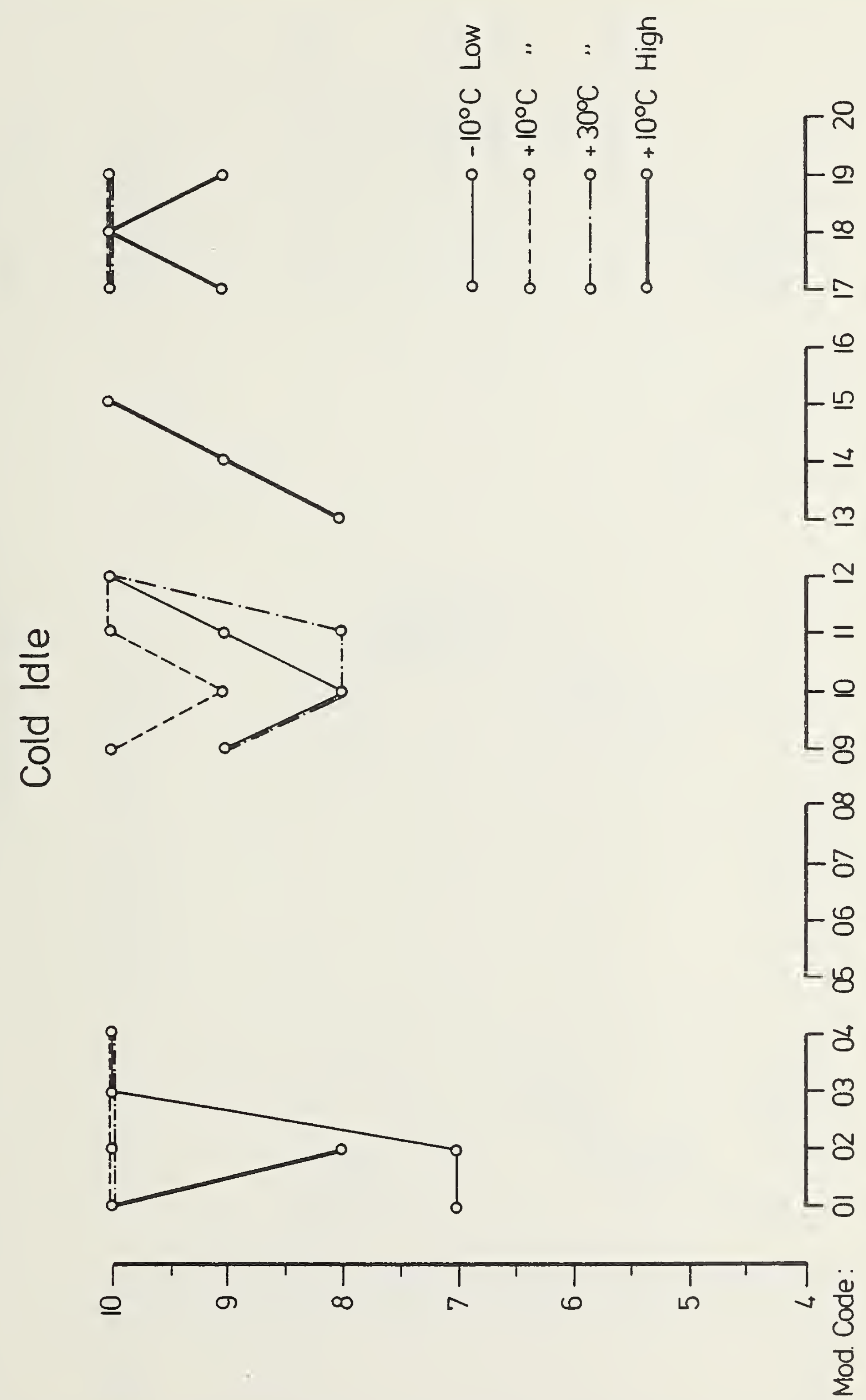

Cold Idle
10
9
8
7
6
5
4
Mod. Code:
01 02 03 04 05 06 07 08 09 10 11 12 13 14 15 16 17 18 19 20
-10°C Low
+10°C "
+30°C "
+10°C High

Fig.: 6. 3. 6

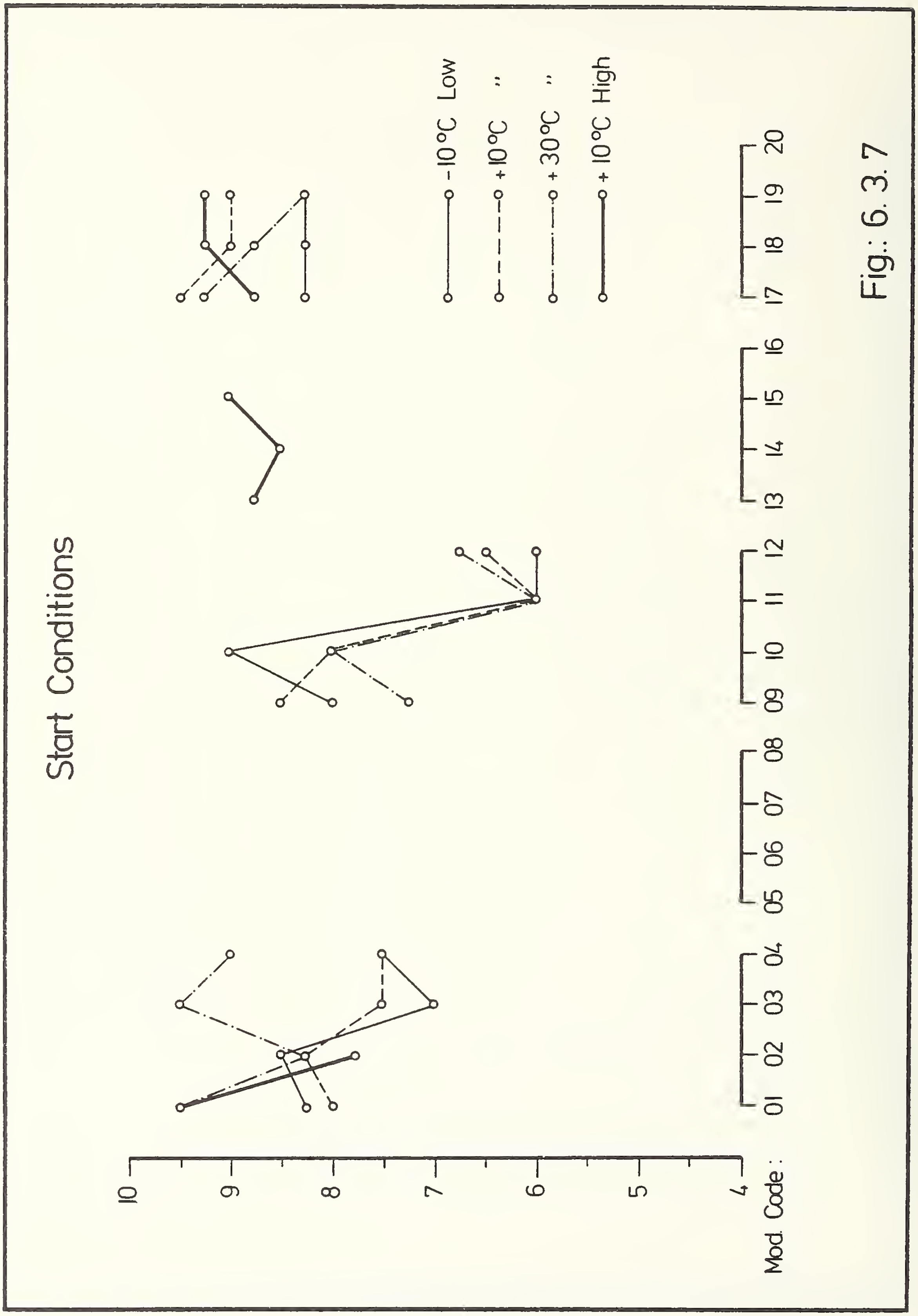

Start Conditions
10
9
8
7
6
5
4
Mod. Code :
01 02 03 04 05 06 07 08 09 10 11 12 13 14 15 16 17 18 19 20
-10°C Low
+10°C "
+30°C "
+10°C High

Fig.: 6.3.7

I. Warm up Period

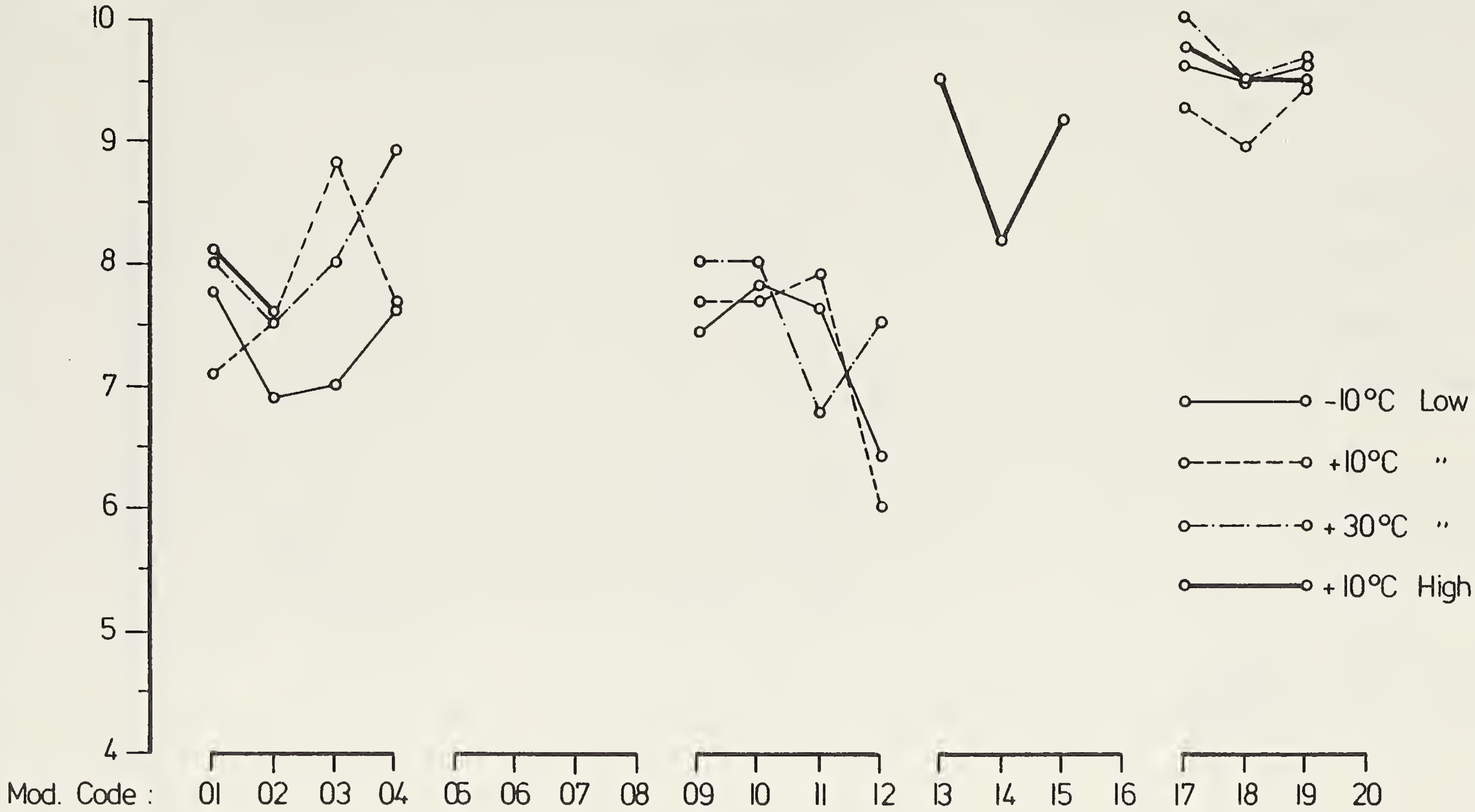


Fig.: 6. 3.8

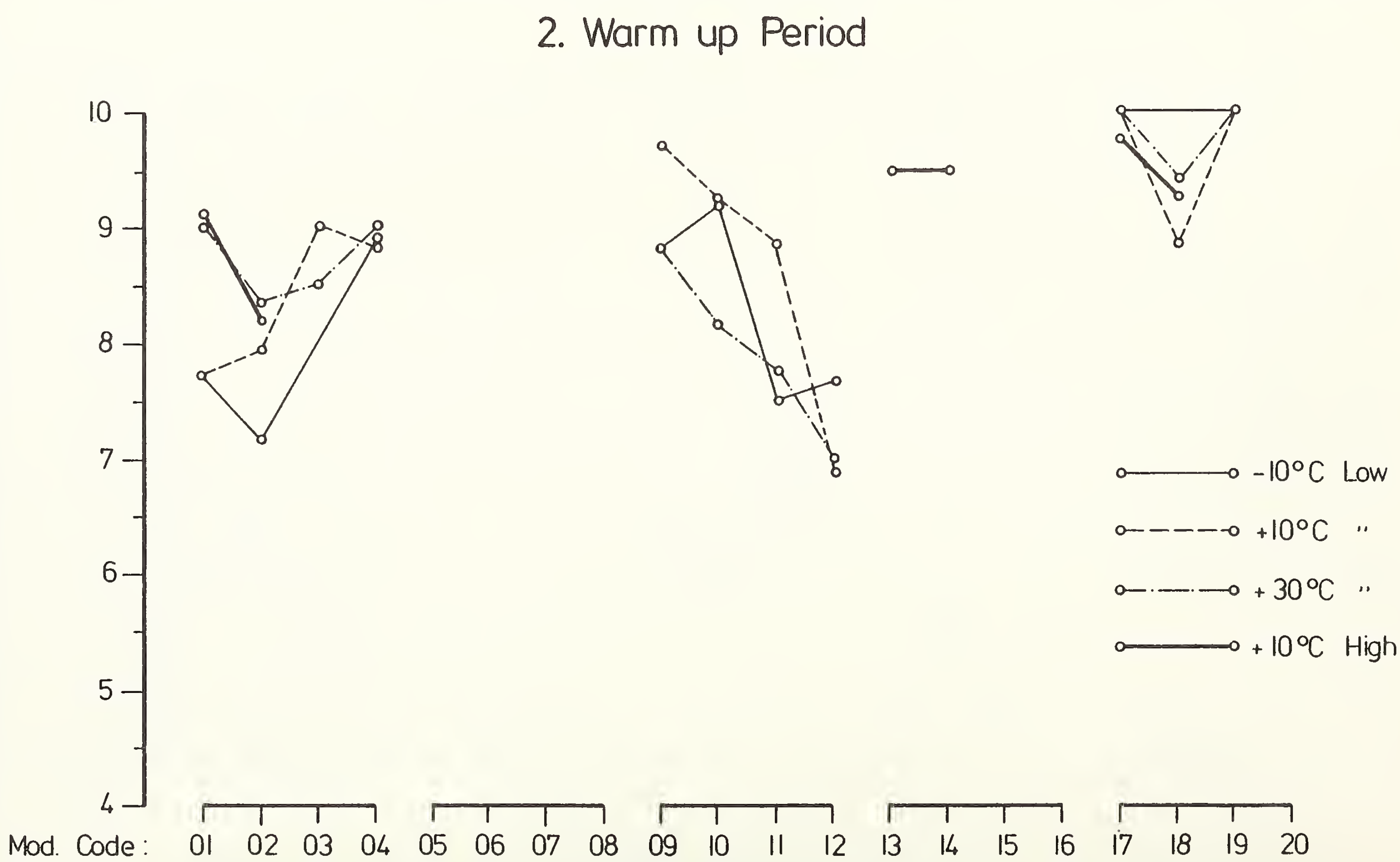

2. Warm up Period
10
9
8
7
6
5
4
Mod. Code :
01
02
03
04
05
06
07
08
09
10
11
12
13
14
15
16
17
18
19
20
-10°C Low
+10°C "
+ 30°C "
+ 10°C High

Fig.: 6. 3.9

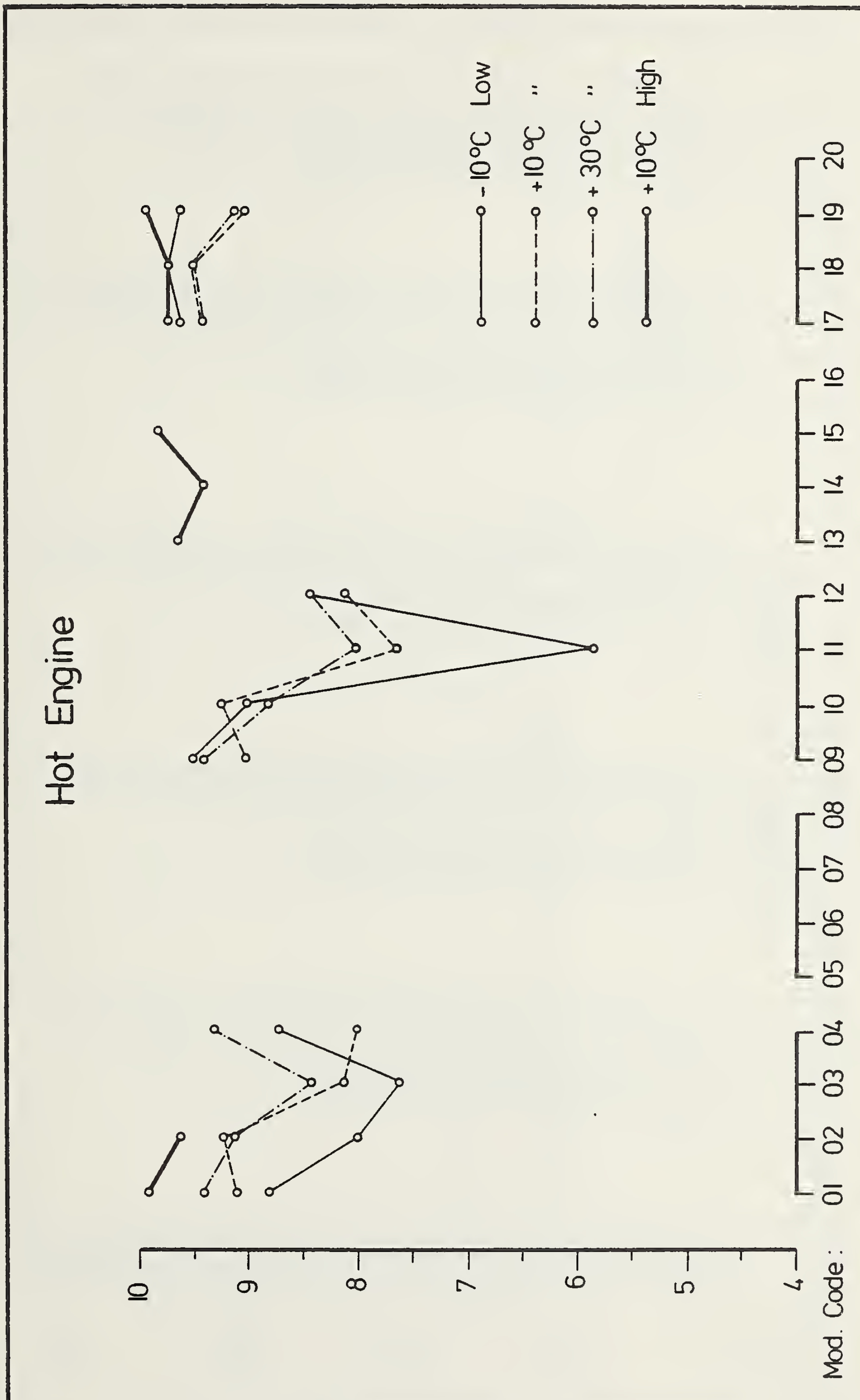

Hot Engine
10
9
8
7
6
5
4
Mod. Code :
01
02
03
04
05
06
07
08
09
10
11
12
13
14
15
16
17
18
19
20
-10°C Low
+10°C "
+30°C "
+10°C High

Fig.: 6.3.10

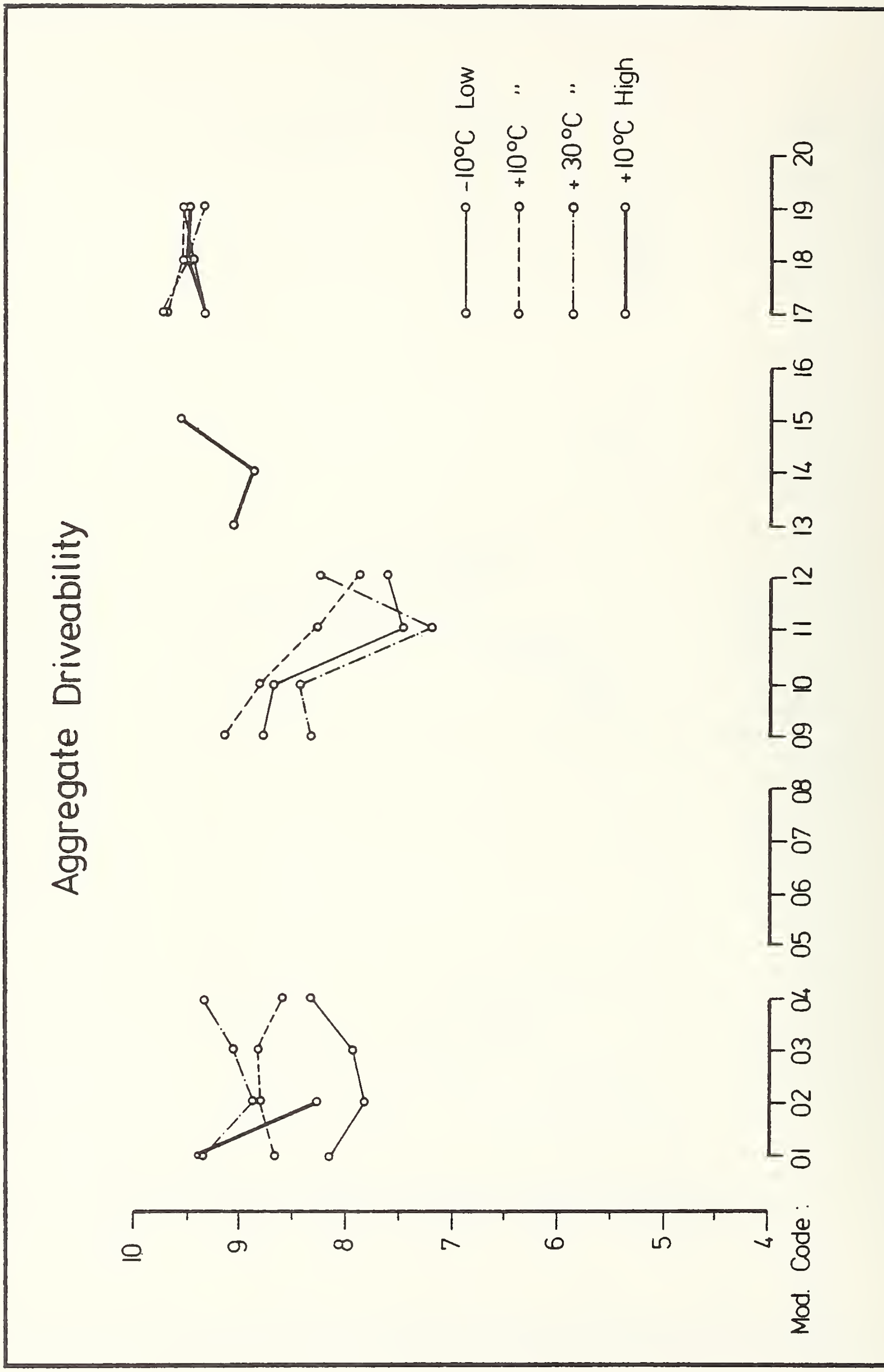

Aggregate Driveability
10
9
8
7
6
5
4
Mod. Code :
01 02 03 04 05 06 07 08 09 10 11 12 13 14 15 16 17 18 19 20
-10°C Low
+10°C "
+ 30°C "
+10°C High

Fig.: 6. 3.11

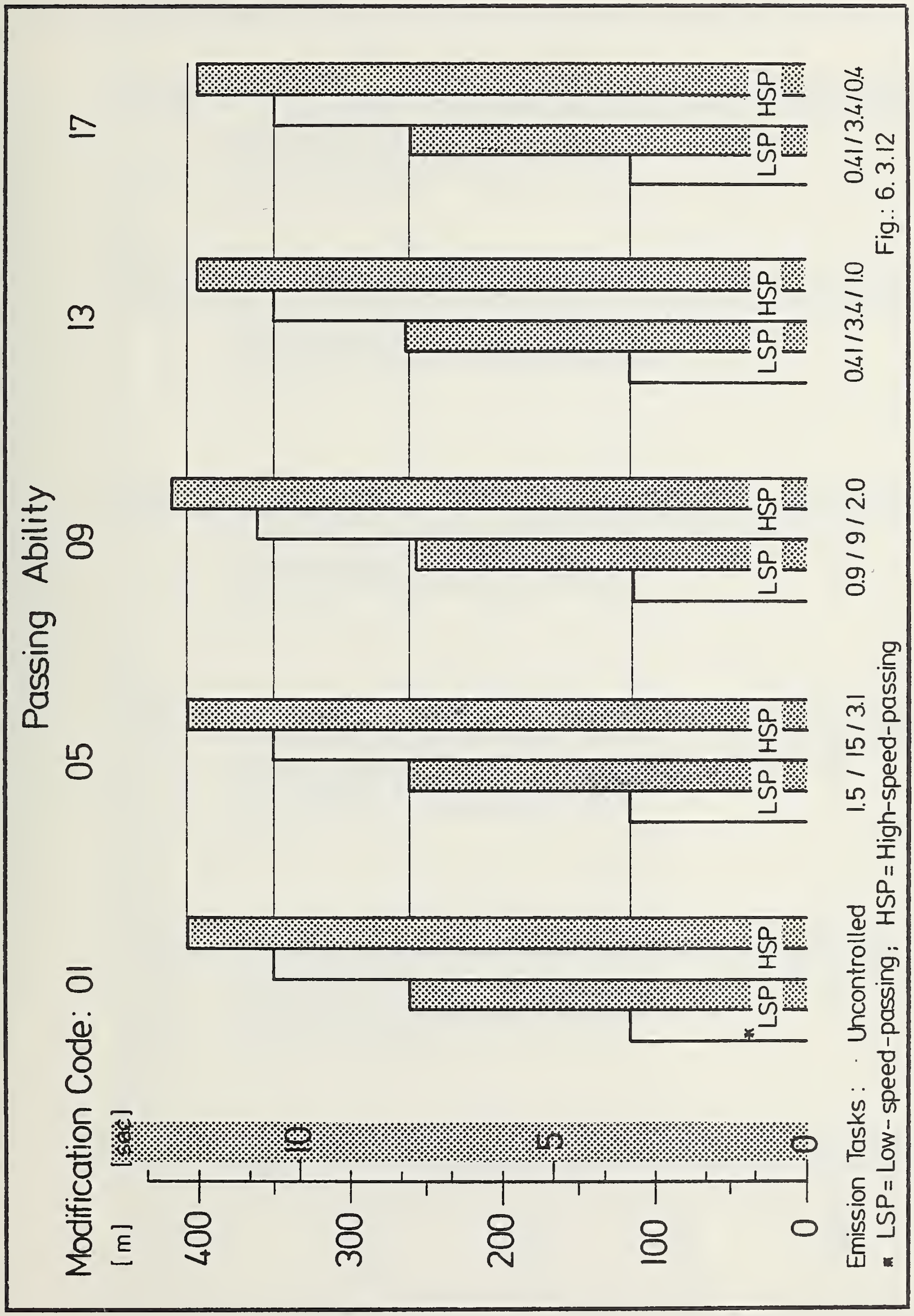
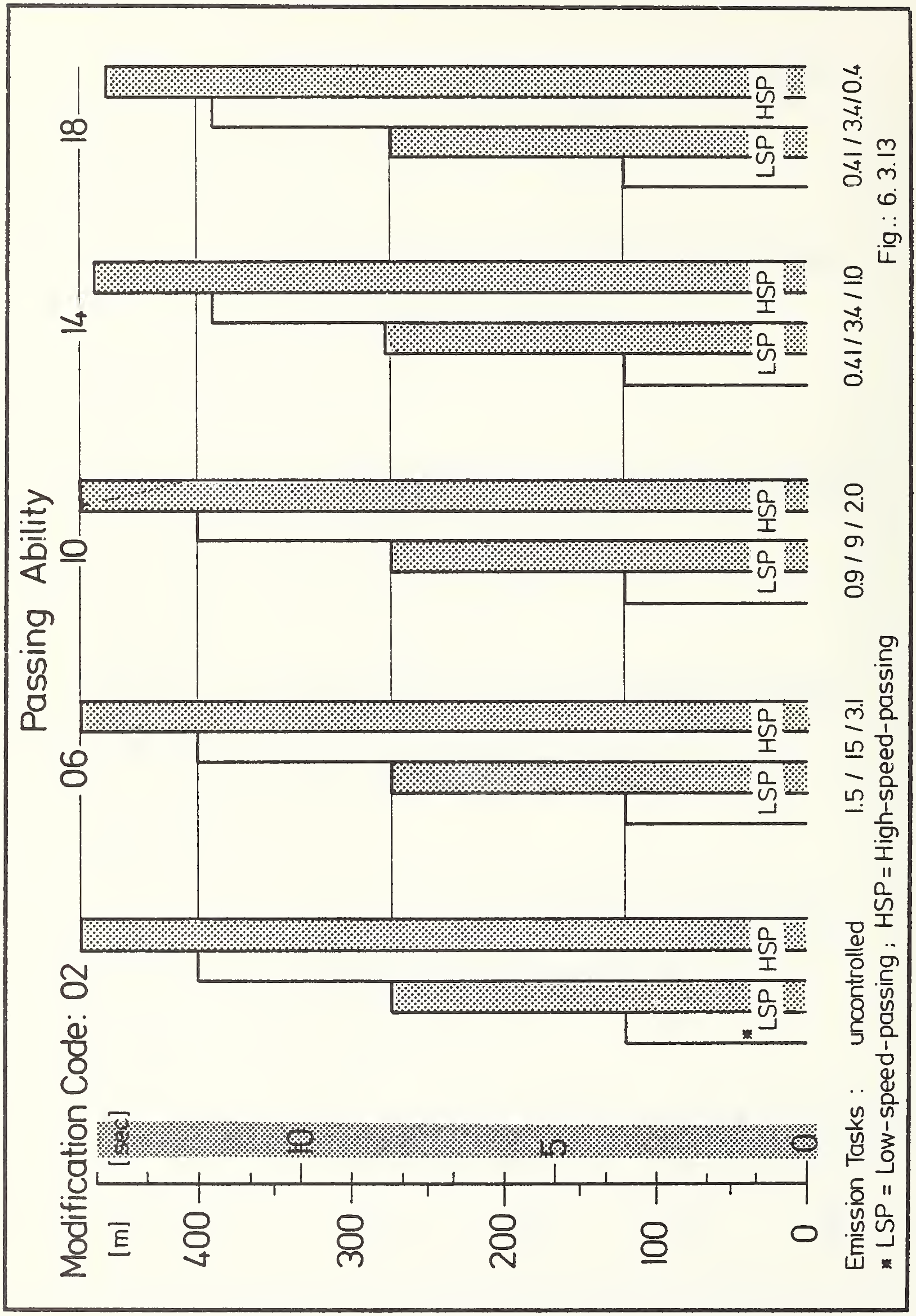
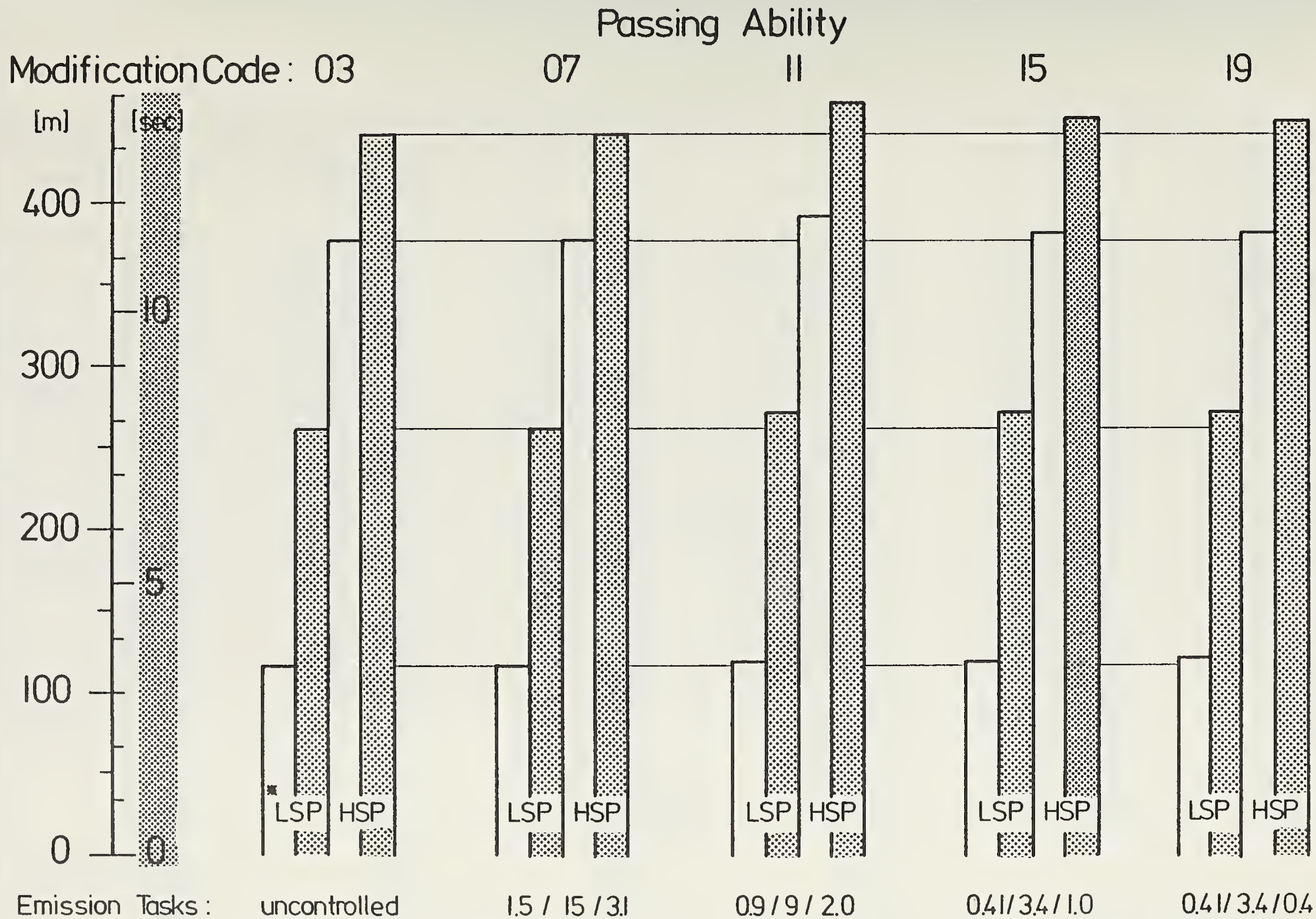

Passing Ability
Modification Code: 01
05
09
13
17
[m]
400
300
200
100
0
[sec]
10
5
0
LSP
HSP
Emission Tasks:
Uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4
* LSP = Low-speed-passing; HSP = High-speed-passing

Fig.: 6.3.12



Passing Ability
Modification Code: 02
06
10
14
18
[m]
400
300
200
100
0
[sec]
10
5
0
* LSP
HSP
LSP
HSP
Emission Tasks :
uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4/0.4
* LSP = Low-speed-passing ; HSP = High-speed-passing

Fig.: 6.3.13



* LSP = Low-speed-passing ; HSP = High-speed-passing

Fig.: 6. 3. 14

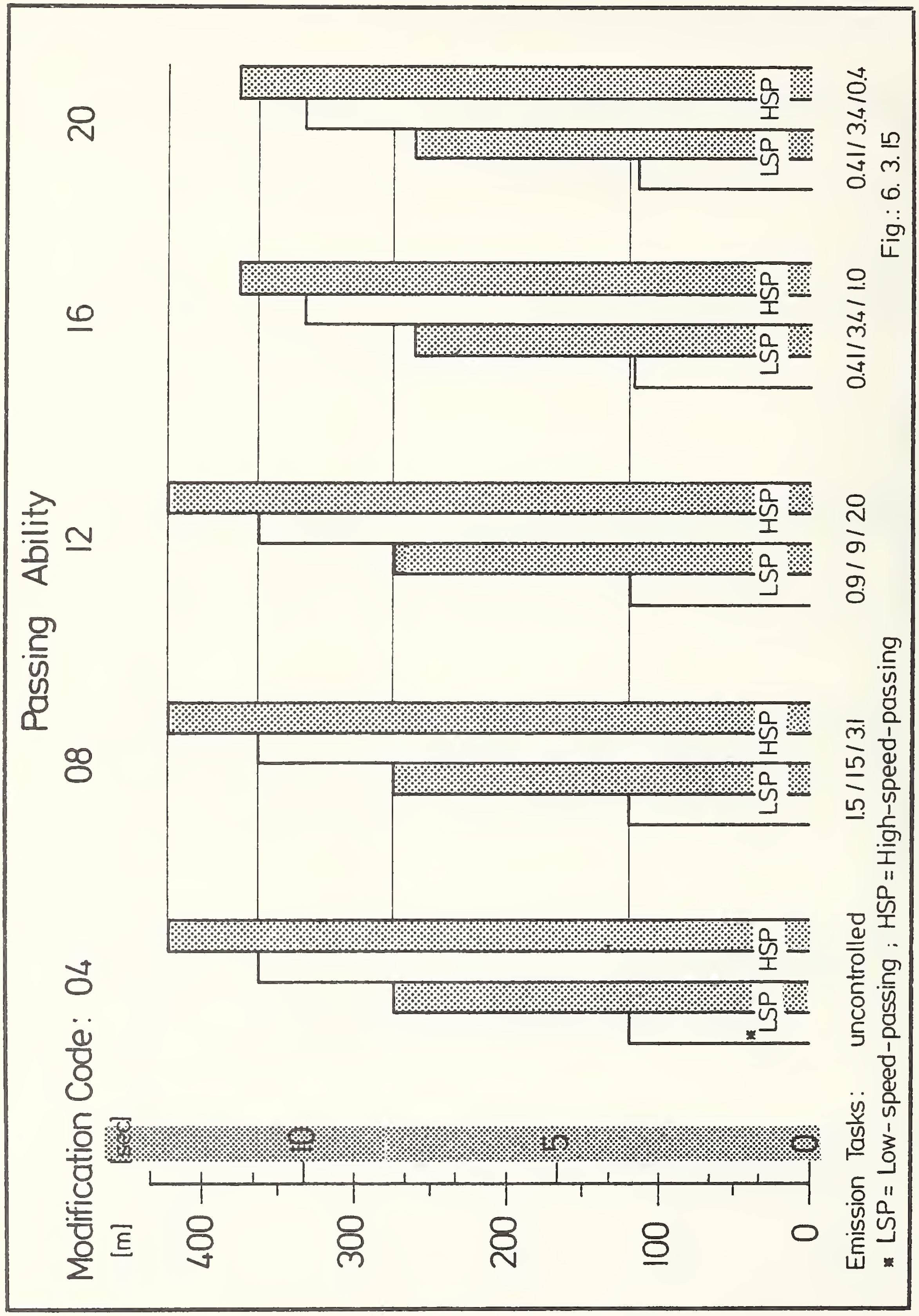

Modification Code : 04
08
12
16
20
Passing Ability
[m]
400
300
200
100
0
[sec]
10
5
0
LSP
HSP
Emission Tasks : uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4
* LSP = Low-speed-passing ; HSP = High-speed-passing

Fig.: 6. 3.15

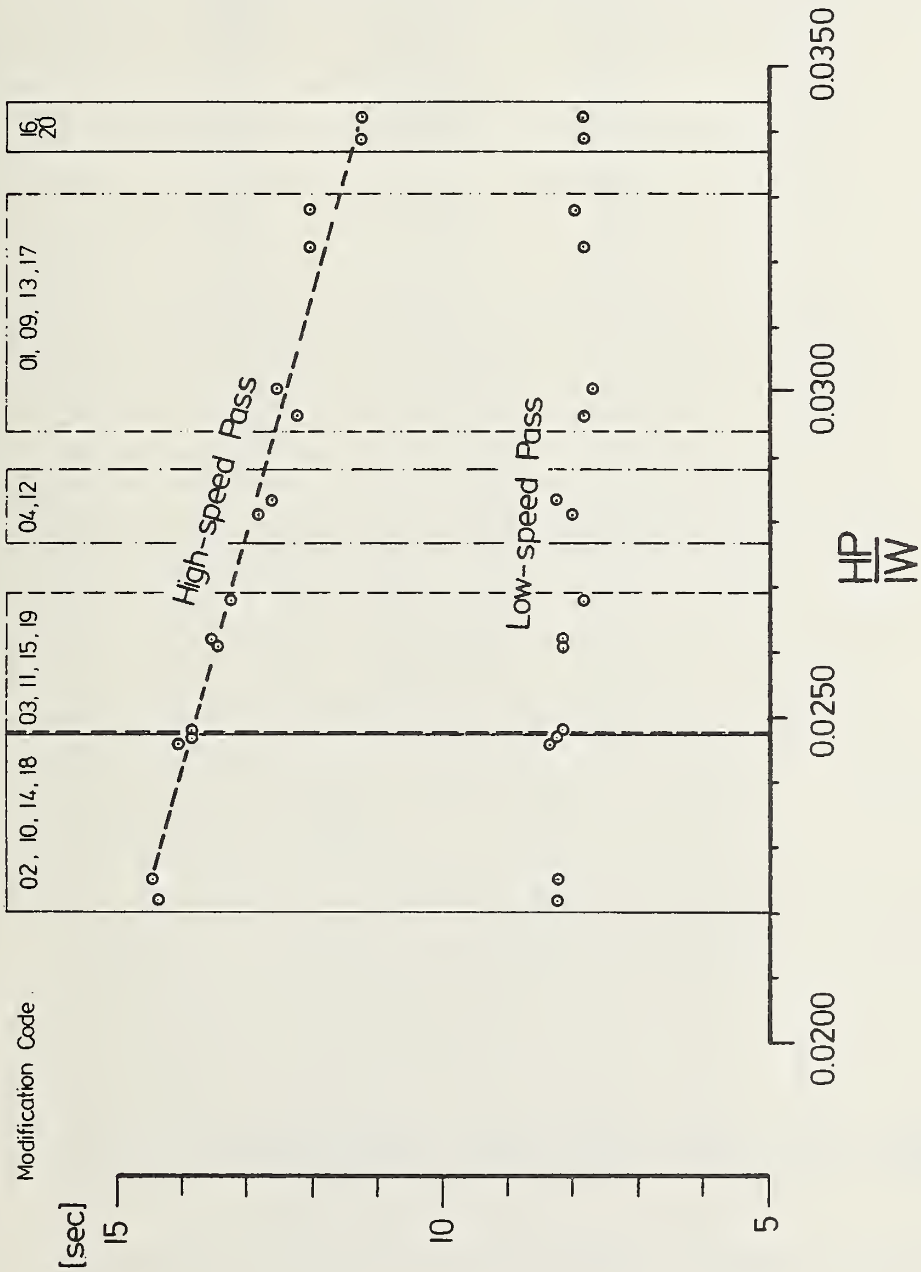
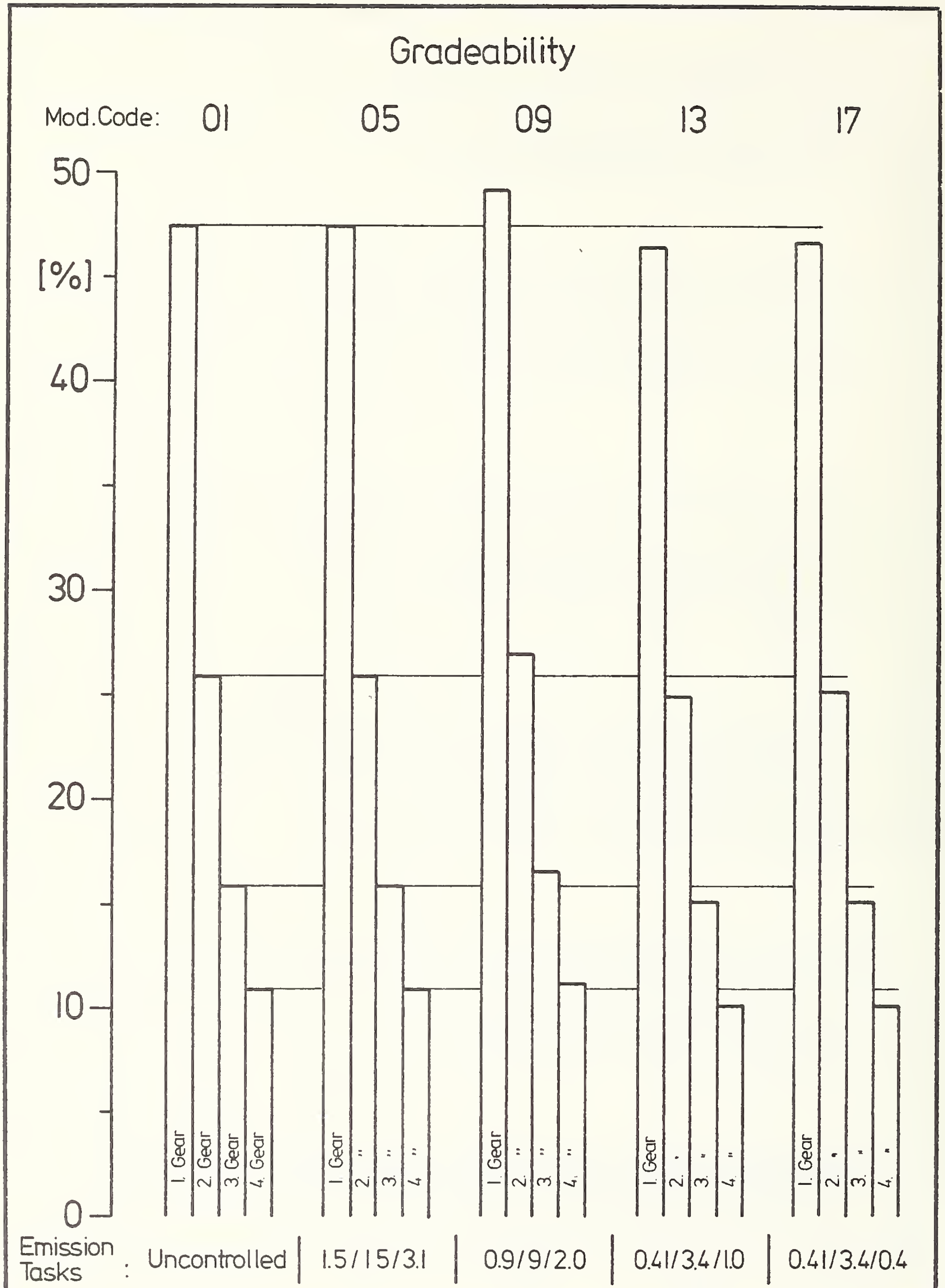

[sec]
15
10
5
Modification Code
02, 10, 14, 18
03, 11, 15, 19
04,12
01, 09, 13, 17
16
20
High-speed Pass
Low-speed Pass
0.0200
0.0250
0.0300
0.0350
HP
IW

Fig.: 6. 3. 16



Gradeability
Mod.Code: 01 05 09 13 17
50
[%]
40
30
20
10
0
1. Gear
2. Gear
3. Gear
4. Gear
Emission Tasks : Uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 6. 3. 17


Gradeability
Mod. Code: 02 06 10 14 18
50
[%]
40
30
20
10
0
1. Gear
2. "
3. "
4. "
Emission Tasks : Uncontrolled | 1.5 / 15 / 3.1 | 0.9 / 9 / 2.0 | 0.41 / 3.4 / 1.0 | 0.41 / 3.4 / 0.4

Fig.: 6. 3. 18

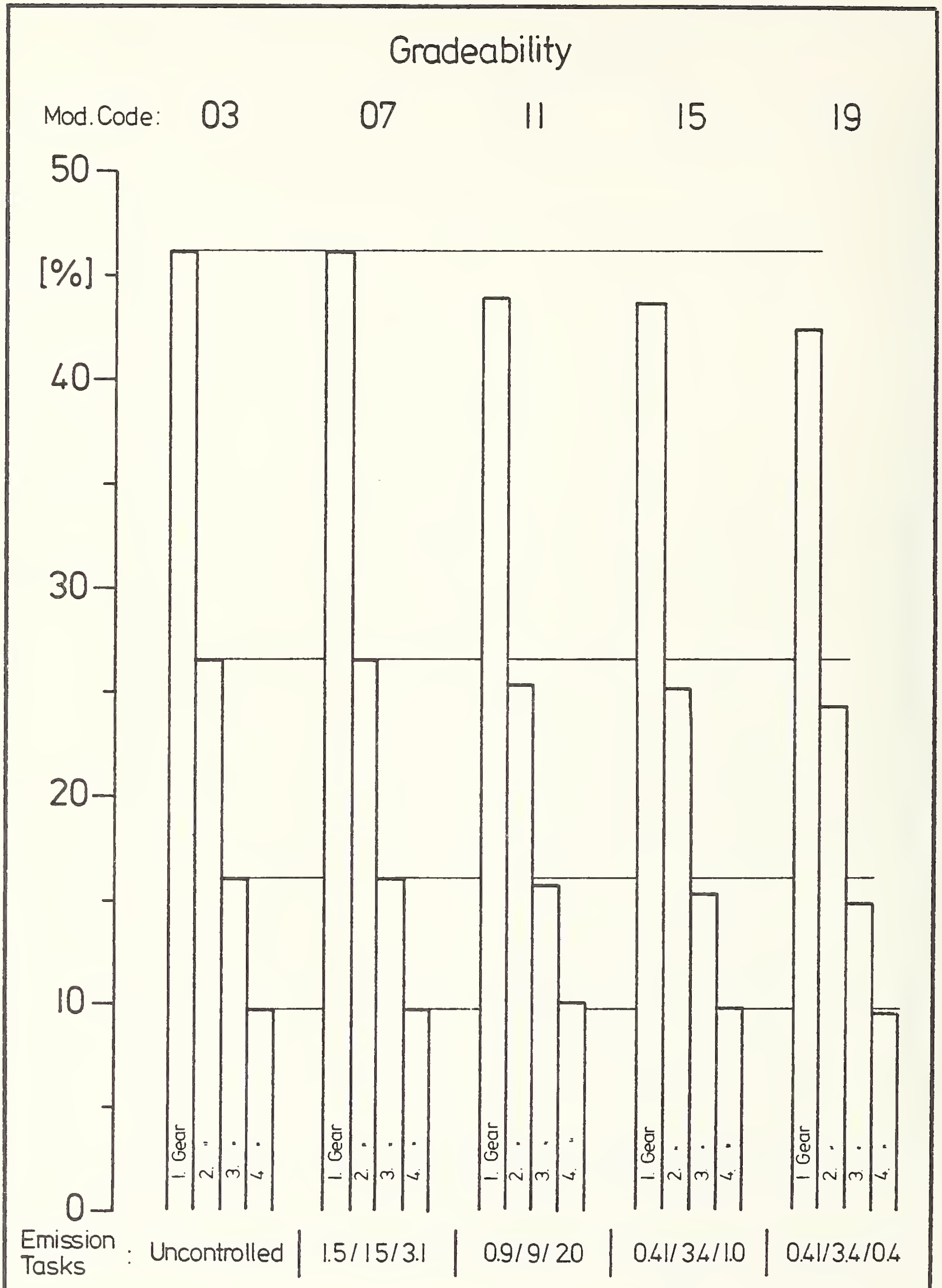

Gradeability
Mod. Code: 03 07 11 15 19
50
[%]
40
30
20
10
0
1. Gear
2. "
3. "
4. "
Emission Tasks : Uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 6. 3. 19

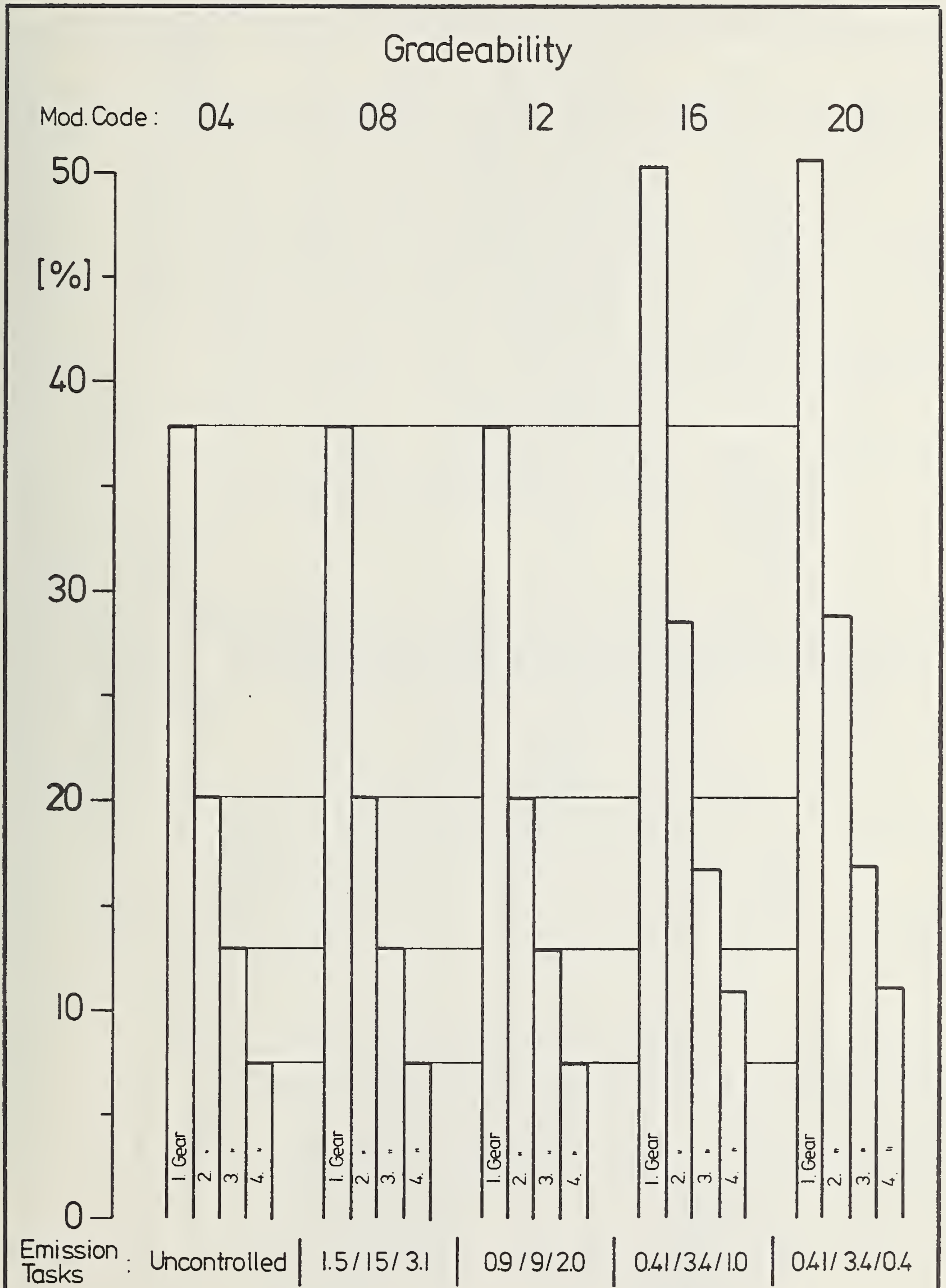

Gradeability
Mod. Code :
04
08
12
16
20
50
[%]
40
30
20
10
0
1. Gear
2. "
3. "
4. "
Emission Tasks :
Uncontrolled
1.5 / 15 / 3.1
0.9 / 9 / 2.0
0.41 / 3.4 / 1.0
0.41 / 3.4 / 0.4

Fig.: 6. 3. 20

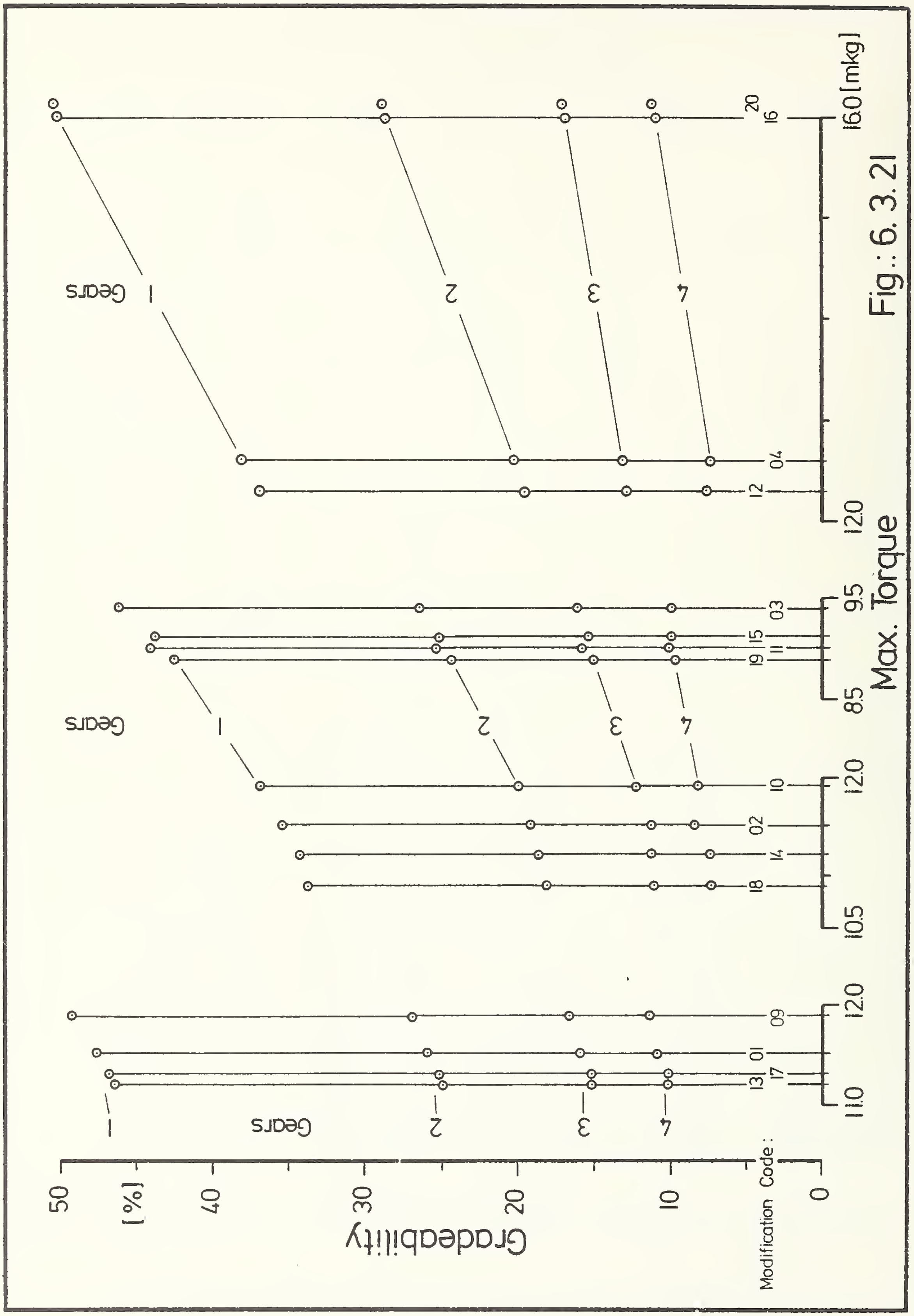

Gradeability
[%]
50
40
30
20
10
0
Modification Code :
Gears
1
2
3
4
13
17
01
09
11.0
12.0
18
14
02
10
10.5
12.0
19
11
15
03
8.5
9.5
12
04
16
20
12.0
16.0 [mkg]
Max. Torque

Fig.: 6. 3. 21

VW Evaluation System										
	Insufficient					Sufficient				
Valuation by Points	1	2	3	4	5	6	7	8	9	10
Nonconformity Assessment	Safety Risk	bad		disagree-able	rework	sufficient	satis-factory, slight Noncon-formities	good, very slight Noncon-formities	very good, hardly noticeable Noncon-formities	excellent no Noncon-formities
	obvious to all Customers	obvious to the average Customer			obvious to the critical Customer			noticeable only to Specialists		not
	continuous	clearly reproducible		frequent		sporadic		hardly reproducible		nonexistent

Tab.: 6. 3.1

Driveability Test	No. 20	Temperature: +10°C	Altitude: low				
Eng. Fam.	Mod. code	License Plate	Emission Task HC | CO | NOx	Displace-ment	HP	IW	HP / IW
A	01	WOB-VA 67	Uncontrolled	96.9	66.7	2 250	0.0296

Cycle No.	Driving Conditions	Assess-ment	AV	Engine Conditions	Weight	Result
1	Starting	10	10	Starting	10%	1.0
2	Cold Idle	10	10	Cold Idle	20%	2.0
3	1. Acceleration	7	8	Start Conditions	20%	1.6
4	Bucking	9				
5	2. Acceleration	7	7.1	1. Warm up Period	15%	1.07
6	3. "	7				
7	4. "	7				
8	5. "	7				
9	6. "	7.5				
10	7. "	7.5	7.71	2. Warm up Period	15%	1.16
11	8. "	7.5				
12	9. "	7.5				
13	10. "	7.5				
14	11. "	8				
15	12. "	8				
16	13. "	8				
17	14. "					
18	15. "					
19	16. "					
20	3 rd gear Acceleration from coasting	9.5	9.1	Hot Engine	20%	1.82
21	2 nd gear Acceleration from coasting	9.5				
22	2 nd gear sudden Acceleration	8				
23	3 rd gear sudden Acceleration	10				
24	last Acceleration	8.5				
	Tab.: 6. 3. 2				Sum	8.65

Modification Code	Startability*			
	Low Altitude			High
	- 10°C	+ 10°C	+ 30°C	+ 10°C
01	10.0	10.0	10.0	10.0
02	10.0	10.0	10.0	8.0
03	9.0	10.0	10.0	
04	6.0	10.0	10.0	
05	10.0	10.0	6.5	
06	9.0	10.0	10.0	
07	10.0	10.0	6.0	
08	6.0	10.0	10.0	
09	10.0	10.0	6.5	
10	9.0	10.0	10.0	
11	10.0	10.0	6.0	
12	6.0	10.0	10.0	
13				9.0
14				10.0
15				10.0
16				
17	8.0	10.0	10.0	9.0
18	10.0	10.0	10.0	9.0
19	10.0	10.0	9.0	9.0
20				

* Best Value 10; Sufficient 6

Tab.: 6. 3.3

Modification Code	Driveability in *																				Driveability			
	Cold Idle				Start Conditions				1. Warm up Period				2. Warm up Period				Hot Engine				Aggregate			
	Low Altitude			High	Low Altitude			High	Low Altitude			High	Low Altitude			High	Low Altitude			High	Low Altitude			High
	-10°C	+10°C	+30°C	+10°C	-10°C	+10°C	+30°C	+10°C	-10°C	+10°C	+30°C	+10°C	-10°C	+10°C	+30°C	+10°C	-10°C	+10°C	+30°C	+10°C	-10°C	+10°C	+30°C	+10°C
01	7.0	10.0	10.0	10.0	8.25	8.00	9.50	9.50	7.75	7.10	8.00	8.10	7.71	7.71	9.00	9.10	8.80	9.10	9.40	9.9	8.13	8.65	9.33	9.37
02	7.0	10.0	10.0	8.0	8.50	8.25	8.25	7.75	6.90	7.50	7.50	7.58	7.17	7.93	8.33	8.17	8.00	9.20	9.10	9.6	7.82	8.81	8.85	8.24
03	10.0	10.0	10.0		7.00	7.50	9.50		7.00	8.80	8.00			9.00	8.50		7.60	8.10	8.40		7.92	8.79	9.06	
04	10.0	10.0	10.0		7.50	7.50	9.00		7.58	7.67	8.88		8.89	8.82	9.00		8.70	8.00	9.30		8.31	8.57	9.34	
09	9.0	10.0	9.0		8.00	8.50	7.25		7.43	7.67	8.00		8.80	9.70	8.80		9.50	9.00	9.40		8.73	9.11	8.30	
10	8.0	9.0	8.0		9.00	8.00	8.00		7.81	7.67	8.00		9.17	9.25	8.13		9.00	9.20	8.80		8.65	8.78	8.38	
11	9.0	10.0	8.0		6.00	6.00	6.00		7.58	7.88	6.75		7.50	8.83	7.75		5.80	7.60	8.00		7.43	8.23	7.17	
12	10.0	10.0	10.0		6.00	6.50	6.75		6.40	6.00	7.50		7.64	6.86	7.00		8.40	8.10	8.40		7.59	7.85	8.21	
13				8.0				8.75				9.50				9.50				9.6				9.03
14				9.0				8.50				8.17				9.50				9.4				8.83
15				10.0				9.00				9.13								9.8				9.56
16																								
17	10.0	10.0	10.0	9.0	8.25	9.50	9.25	8.75	9.59	9.25	10.0	9.75	10.0	10.0	9.75	9.75	9.60	9.40	9.40	9.7	9.31	9.67	9.69	9.31
18	10.0	10.0	10.0	10.0	8.25	9.00	8.75	9.25	9.45	8.92	9.5	9.50		8.83	9.40	9.25	9.70	9.50	9.50	9.7	9.43	9.52	9.49	9.51
19	10.0	10.0	10.0	9.0	8.25	9.00	8.25	9.25	9.57	9.42	9.67	9.50	10.0	10.0	10.0	10.0	9.60	9.00	9.10	9.9	9.51	9.51	9.32	9.46
20																								

* Best Value 10; Sufficient 6

Tab.: 6. 3. 4

Modification Code	Acceleration Performance				Gradeability [%]				Performance [HP]
	Low-speed Pass		High-speed Pass						
	[mi]	[sec]	[mi]	[sec]	1. gear	2. gear	3. gear	4. gear	
01	0.071	7.8	0.217	12.2	47.5	25.8	15.8	10.8	66.7
02	0.073	8.2	0.248	14.3	35.4	19.0	11.5	7.9	"
03	0.071	7.8	0.232	13.2	46.0	26.3	15.9	9.7	60.4
04	0.073	8.2	0.223	12.6	37.8	20.1	12.8	7.3	84.9
09	0.070	7.65	0.223	12.5	49.2	26.8	16.5	11.1	67.6
10	0.073	8.2	0.248	14.4	36.7	19.8	12.1	8.1	"
11	0.072	8.1	0.242	13.8	43.9	25.3	15.6	10.0	55.8
12	0.072	8.0	0.226	12.8	36.7	19.4	12.6	7.4	88.5
13	0.071	7.9	0.217	12.0	46.3	24.9	15.1	10.1	73.8
14	0.073	8.3	0.242	14.0	34.2	18.4	11.2	7.5	"
15	0.072	8.1	0.236	13.5	43.6	25.1	15.3	9.7	58.9
16	0.071	7.8	0.204	11.2	50.1	28.5	16.7	10.8	101.7
17	0.070	7.8	0.217	12.0	46.6	25.1	15.1	10.1	72.5
18	0.073	8.2	0.242	13.8	33.6	18.0	10.9	7.3	74.1
19	0.073	8.1	0.236	13.4	42.4	24.3	14.9	9.5	58.8
20	0.070	7.8	0.204	11.2	50.4	28.7	16.8	11.0	102.6

This are values of single cars without cowsideration of production spread

Tab.: 6. 3. 5

Modification Codes	Prices Related to Modification 01	02	03	04
01	100			
02	100	100		
03	89		100	
04	110			100
05	118			
06	121	121		
07	106		120	
08	131			119
09	121			
10	121	121		
11	110		124	
12	131			119
13/17	178			
14/18	189	189		
15/19	146		164	
16/20	196			177

Tab.: 6. 3. 6

BIBLIOGRAPHY

(1) Klingenberg, H. u.a.: Prüfmethoden und Meßverfahren für Automobilabgase. VW Forschungsbericht 1977

(2) Hurter, D.A. et.al.: A Study of Technological Improvements in Automobile Fuel Consumption. Final Report 1974 U.S. Department of Transportation and U.S. Environmental Protection Agency.

(3) Hucho, W.-H. et.al.: The Optimization of Body Details - A Method for Reducing the Aerodynamic Drag of Road Vehicles. SAE Paper 760 185, 1976

(4) Oetting, H.: Beitrag zur Erscheinung des Klopfens in Ottomotoren. Dissertation Braunschweig 1968.

(5) Derndinger, H.-O. und Ritter, H.: Über den Einfluß des Hubbohrungsverhältnisses. MTZ 27, 322-325 (1966).

(6) Borrmeister, J.: Fremdgezündeter Vergasermotor mit Kolbenbrennraum und Ladungsdrall. Kraftfahrzeugtechnik 6, 171-174 (1973).

(7) Arama, C.: New Approaches to Fuel Economy in Spark-Ignition Engines. Journ. Aut.Eng. 11-14 (June 1975).

(8) Huber, E.: Untersuchungen über das Verfahren der Verlängerten Dehnung. Dt. Kraftfahrtforsch. und Straßenverkehrstech. 95 (1956).

(9) Schweitzer, H. und Zeilinger, K.: Das Optimale Luft-Brennstoff-Verhältnis für Ottomotoren. MTZ 32, 16-21 (1971).

(10) Choné, G. und Oetting, H.: What can we do to decrease NO_x with rich mixtures? 14th Fisita Congress London 1972

(11) Lies, K.H. et.al.: HCN Exhaust Gas Emissions of Gasoline Engines. VW-Research Report No. MT-F5-77/8

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